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Port Hueneme, California 93043-4370

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CR 00-002-AMP

CONCEPTUAL DESIGN OF AN AUTONOMOUS MARINE BOOSTER PUMP

An Investigation Conducted by:

Howland Associates
Danville, New Hampshire

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13. ABSTRACT *(Maximum 200 words)*

The objective of this phase of the program was to develop a detailed design for a prototype autonomous marine booster pump (AMBP). Principal tasks included a mechanical design of the complete system including the mechanical, hydraulic, and electrical components and a hull structure to provide flotation. The system was analyzed to determine the mooring loads and a mooring system was designed. The major remote control and telemetry system components were procured and bench tests were run to program and test simulated operations. These were also tested with radio modem communication between the shore and on-board systems to simulate the actual field situation and to determine the attainable range. It was concluded that the AMBP is still feasible and should be carried into the prototype testing phase, where several important parameters will be established for the final system design.

14. SUBJECT TERMS

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PREFACE

The authors would like to express their appreciation for the opportunity to work on this interesting project. It is certainly one of the more interesting, and challenging, projects with its many varied technologies and tasks.

We have been helped by many individuals who were not on the project team. In particular, we would like to acknowledge the help and guidance of Mr. Laurence G. "Chip" Nixon of the Naval Facilities Engineering Command. We were aided by a number of individuals among the component manufacturers and others with no involvement in the project. In particular, Mr. John C. Thomsen of Cummins Northeast and Mr. William M Rehor of SEN-DURE Products have responded well beyond the normal call of duty. We were also helped considerably at a critical point in the project by the advice of Mr. Bruce Markel.

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1. INTRODUCTION

1.1 BACKGROUND

For amphibious operations, it is necessary to establish an assault fuel supply system on the beach. The fuel is supplied to storage tanks and fuel dispensing equipment through a hoseline from a supply ship anchored off shore.

In some locations, the available depth, bottom conditions, or tides may force the supply ship to anchor several miles offshore, exceeding the capability of the shipboard pumps to transfer the fuel to the shore installation at high flow rates.

The shipboard pumping capability cannot, in general, be extended by a boost pump located on the beach without collapsing the hoseline or cavitating the inlet of the boost pump. In fact, the optimum location for a boost pump system is in the hoseline approximately midway between the supply ship and the beach.

However, the only self-powered pumps available for this purpose are large trailer-mounted or skid-mounted units which would require a vessel anchored along the hoseline and would have to be tended by an operator. This would increase the manpower requirements for the assault unit and would place the operator in a potentially vulnerable location.

Thus, there is a requirement for a reliable, remotely operated and monitored, self-powered boost pump system. The system must be deployable along the hoseline in its own package or hull and capable of being moored in place. It is also desired that, except for starting up and occasional monitoring by shore personnel, the boost pump be essentially automatic. This system has been named the Autonomous Marine Booster Pump (AMBP).

In Phase I of this project, Howland Associates conducted a study to develop the design requirements for the AMBP and to develop the preliminary conceptual design and layout for the system.. (1)* As a result of Phase I, it was recommended that the conceptual design be carried to the next stage of development with the detailed design, fabrication, and testing of a prototype AMBP. It was also recommended that benchtop experimental development tests be accomplished on the major elements of the remote computer control system which will be used to control the AMBP from the shore.

1.2 OBJECTIVE

The main objective of the Phase II work to date has been to accomplish a detailed design of a prototype AMBP system which will allow fabrication of the unit for testing.

An additional objective has been to obtain the major hardware and software components of the remote control system for the AMBP and to program these units to simulate the control tasks which will be required for the AMBP. The goal was to verify their capability to perform the control functions over ranges typical of the ultimate application.

*Numbers in parentheses refer to references listed at the end of this report.

1.3 SCOPE OF WORK

In order to accomplish these objectives, this project has included the following major task areas:

- A detailed mechanical design of the AMBP prototype leading to drawings that can be used to build a prototype unit for development testing.

- Design analysis to verify, insofar as possible, the mechanical strength of the prototype unit and its capability to meet the various operational requirements.

- An electrical design of the AMBP prototype system to provide the drawings and data necessary to purchase the electrical hardware and wire the unit.

- Analysis of the roll stability of the AMBP system to ensure that it will be stable under the anticipated operating conditions.

- Analysis of the mooring loads for the AMBP under its anticipated operating conditions and design of the mooring system required to maintain its position.

- Benchtop tests to program and test the remote control hardware to simulate the major control operations and obtain experience with its programming.

- A range test, using the remote control hardware and radio modems, to verify that the control operations can be accomplished at distances comparable to the actual range required of the AMBP control system.

These tasks are discussed in detail in subsequent sections of this report.

2. SUMMARY

A detailed mechanical design was developed for the prototype AMBP system. The design was carried out using the AutoCAD computer aided design program. Major assemblies include the Pump/Pallet Assembly, containing all of the hydraulic hardware and most of the electrical and electronic components and the Hull Assembly, which also includes the engine fuel tanks. The major electrical components and their enclosure was shown in the mechanical design. The electrical design and wiring diagrams for the project were not prepared on a CAD system and have been produced only in hard-copy form.

During the design process, attention was devoted to assessing the feasibility of operating the AMBP in a location where it would be periodically left grounded without a source of seawater cooling. This condition was analyzed, and it was decided to cool the engine, through a heat exchanger, by means of the fuel (or water) being pumped by the AMBP itself. Accordingly, two possible methods of accomplishing this objective were analyzed and it was decided that the optimum method was to draw off a portion of the main flow by means of a special jet-pump fitting and return it to the line after it passes through the heat exchanger. Howland Associates worked with SEN-DURE Products, Inc., the suppliers of heat exchangers for Cummins engines, to design a special heat exchanger for this purpose. A special jet pump fitting was designed for the pumped flow line to supply coolant to this heat exchanger.

Because of size limitations, especially the width or beam of the AMBP, it was clear that there could be little space around the engine, pump, and piping system. Thus, it was decided to design the unit to be capable of easily being disconnected from the hull and lifted out for servicing. This approach has been followed in the design. Periodic checks of fluid levels and replenishment can be accomplished through access hatches. Also, main power can be turned on through these access hatches. After the power is on, the system can be operated from outside the hull using the shore-based terminal or an equivalent PC-based terminal equipped with a radio modem.

Because the AMBP contains relatively heavy equipment inside the hull and layout limitations require that significant portions of this equipment must be relatively high within this envelope, there is a need from a flotation stability standpoint, to provide considerable structural and payload weight near the bottom of the envelope. Thus, it was found desirable to fabricate the hull from steel and to provide a fairly heavy, overdesigned structure near the keel. This approach, of course, results in a fairly high overall weight. It is currently estimated to be about 11,400 lbs. This weight is still well below the capacity of the crane on the side loading warping tug.

The major stresses in the mechanical components have been analyzed to determine conservative values of stress level under the worst loading conditions and all have been found to be safe.

Thermal analyses have been performed to size the electrical enclosure heater for cold weather operation and to predict the compartment temperatures under the worst specified hot weather conditions. The analysis indicates that it is likely that the engine compartment will remain below a safe temperature, even under extreme ambient air conditions and solar radiation. However, the actual heat load rejected by the engine to the surrounding compartment is not known. Thus, some warm weather testing and analysis to establish this heat load should be planned for the prototype testing.

The engine cooling system was also designed on the basis of the design flowrate of 600 GPM at maximum hoseline pressure. At much lower flowrates, the cooling requirements for the engine are expected to decrease considerably, but the cooling available from the fuel flow will also decrease. The actual engine cooling rate required, for example, at idle speed and very low output power is not available. Thus, some testing of the prototype under widely varying flow conditions will be required to establish the adequacy of the cooling system. The jet pump fitting can easily be modified to increase cooling flow by further restricting the throat. This, of course comes at a price of increasing the pressure drop in the main line. This pressure drop, however is currently negligible and can be increased considerably before it creates a problem for the design.

Some design iteration was required to obtain an overall AMBP design which was flotationally stable in roll. At first, the hull was too narrow and the center of mass too high for stability. It was found necessary to increase the beam of the hull to the maximum possible which still allows it to be fitted within an ISO shipping container. Then, aluminum was specified for the hatch material and the batteries and other heavy components were relocated or repositioned to lower the center of mass as much as possible. The final design is stable and would require approximately four men standing on one rim to roll the hull 15 degrees.

After the hull configuration was established, considerable analysis was devoted to determining the requirements of the mooring system. Several candidate designs were examined and the optimum was selected to be a 1-inch nylon mooring line on each end of the hull, assuming that the current can reverse.

The system was analyzed under both a design current of 2.5 knots and a survival current of 5 knots to determine the mooring line and hoseline loads for use in the stress analysis. A pitch stability analysis was then conducted to determine the optimum point of attachment for the mooring lines. This optimum point was determined to be the junction between the upper and lower hull.

A detailed design of the mooring system was then performed and a bill of materials compiled. A single 100 lb. NAVMOOR anchor is specified for 2.5 knots. For 5 knots, a second NAVMOOR anchor is specified along with a 1000 lb. clump or sufficient additional chain to balance the vertical component of line tension, 1370 lbs.

Most of the hardware and the software required to implement the prototype AMBP control system was obtained and set up for bench testing and development. The OPTO 22 Cyrano software was used to program the on-board controller to simulate a pump control operation similar to the one which will actually be required on the AMBP. This was tested and found to operate well. The Mistic MMI software was used to develop a base-station screen for this simulation program as well as prototype screens for displaying the actual telemetry and control data that will be required on the AMBP.

The modems were programmed with the assigned military frequencies and set up with the OPTO controller and a base-station PC for radio communication testing. The OPTO controller was used as a battery-powered mobile unit in a truck. The range between the base station and the mobile unit was then increased while testing the pump control simulation program. Despite the hilly terrain, successful communication at a range up to about 2 miles was established. It is expected that there should be no problem under the much better conditions on the beach.

3. OVERVIEW OF THE DESIGN PROCESS

The major basic design decisions were made during Phase I of this project and are reported in Reference (1) along with their underlying rationale. The design tasks of Phase II reported in this report have been concerned with the following:

- The physical layout of the detailed hardware to assure that the parts will fit together without interference and fit within the overall allowable envelope.
- Detailed drawings from which fabricated parts can be made.
- Analysis of performance factors which could not be addressed until a detailed picture of the hull or other structures was available.
- Obtaining experience with the control and telemetry software to provide a basis for detailed programming later when all of the AMBP hardware is available.

3.1 CAD DESIGN

The detailed mechanical design for Phase II was performed using AutoCAD. Thus, as the design progressed and the details for each part were added, the program was able to ensure that no interferences occurred. The program also provided weight and center of gravity information for the buoyancy and stability analyses of Sections 6. and 7.

Fairly major design modifications were made as the design process proceeded but the use of the AutoCAD system allowed these design changes without a large amount of drawing revision such as would have been required for paper drawings.

The final design is shown in Figure 3.1 which shows an exploded view of the AMPB hull, hatch and the interior machinery pallet. Figures 3.2 and 3.3 show the machinery pallet in more detail.

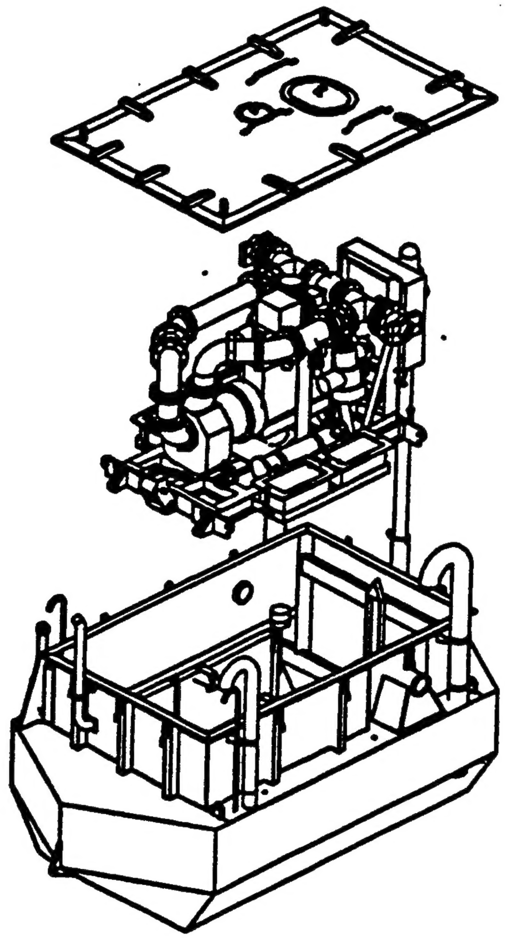
A complete set of mechanical drawings for the AMBP have been provided separately in electronic and hard-copy form. Electrical wiring diagrams, which were not prepared on a CAD system, have been provided in hard copy only.

3.2 DESIGN CRITERIA

The major design criteria were developed in Phase I and reported in Reference (1). However, during Phase II a major revision in the design criteria was made and several criteria were refined. These are discused as follows:

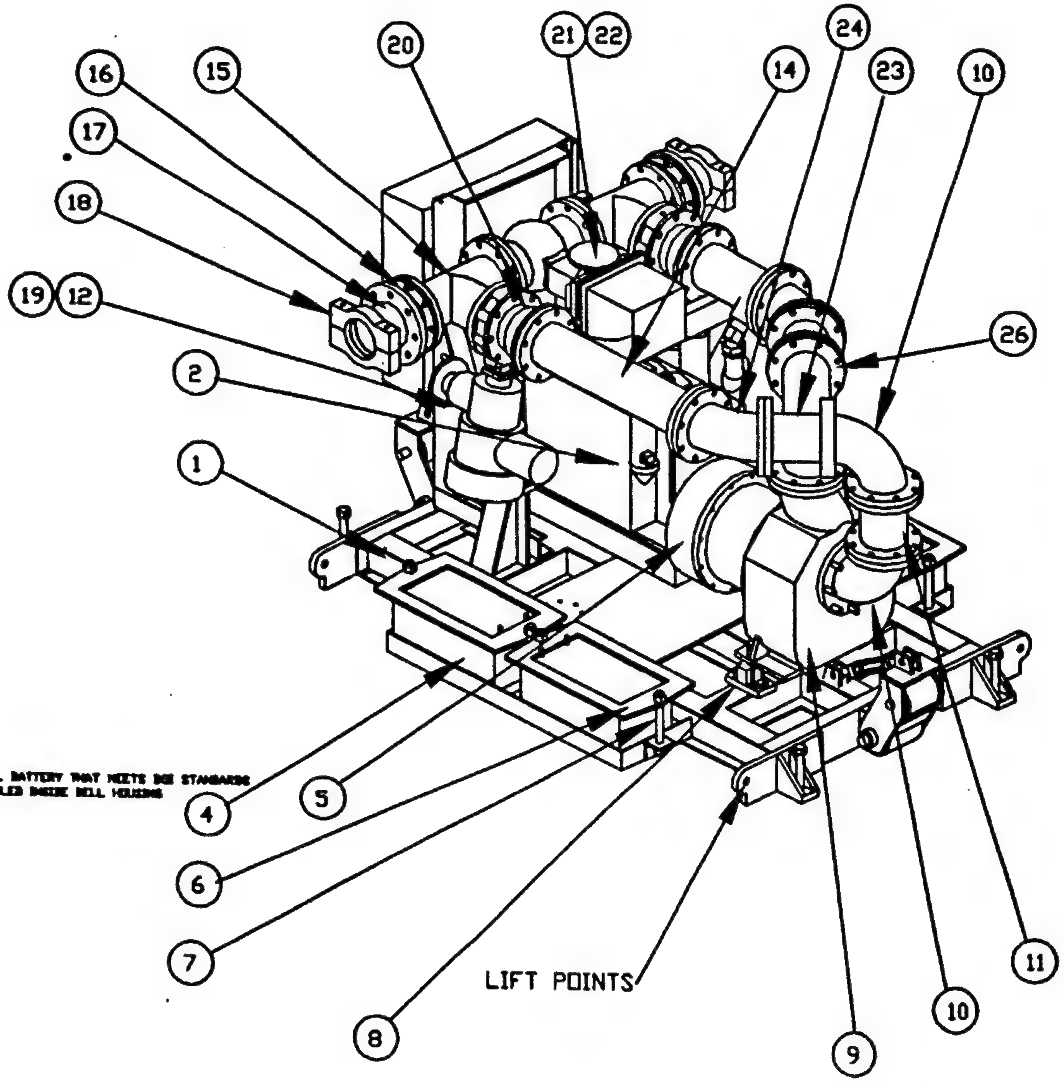
(a) Engine Cooling

During Phase I, it was assumed that the marine diesel engine, used to power the pump, would be cooled by means of a heat exchanger, using seawater as the secondary or final coolant.



Exploded View of the AMBP

Figure 3.1



NOTES
(1) USE ANY COMMERCIAL BATTERY THAT MEETS BCI STANDARDS
(2) COUPLING IS ASSEMBLED INSIDE BELL HOUSING

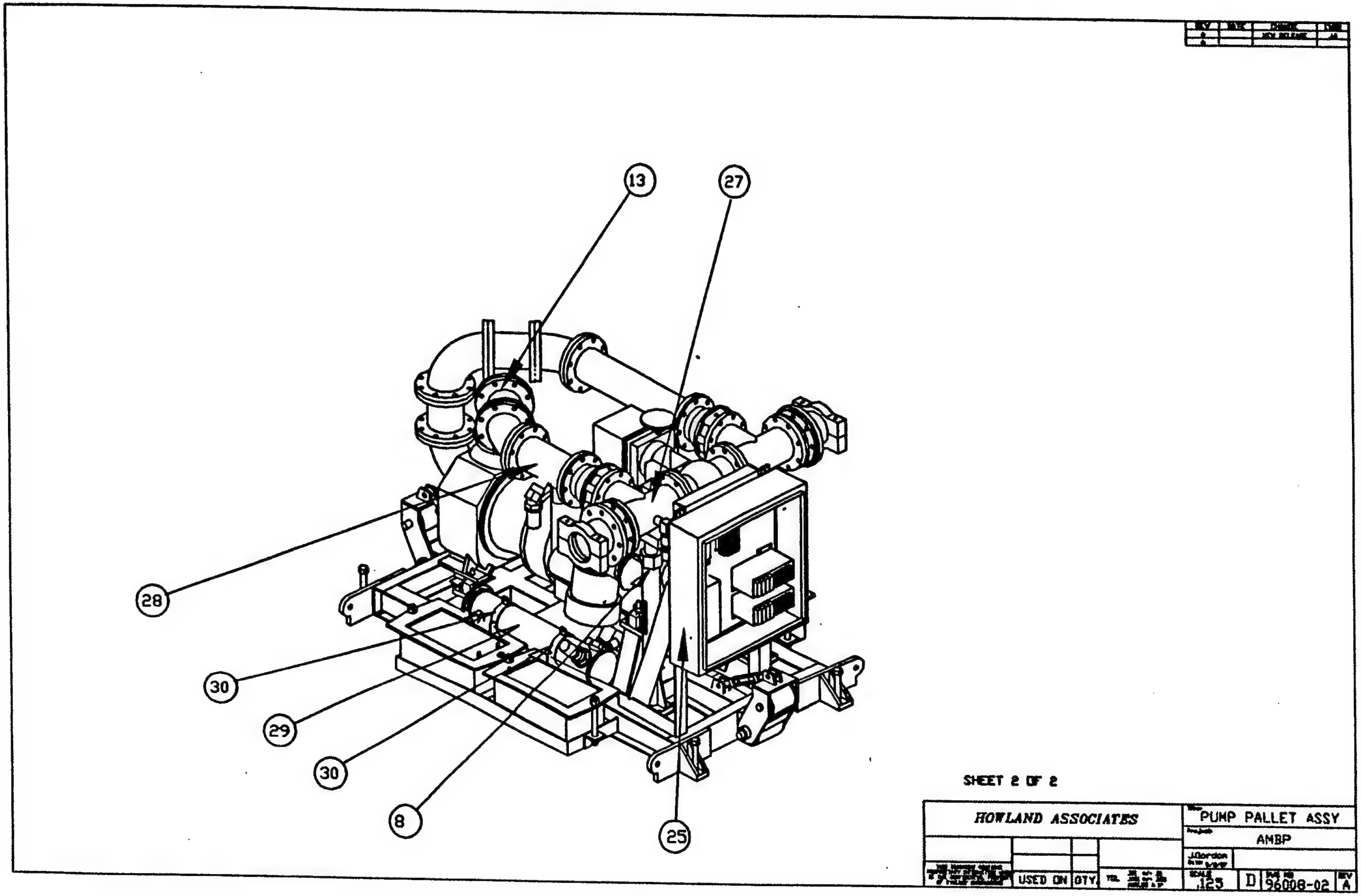
ITEM	QTY	DWG#	DESCRIPTION
33	20	NG60	DIXON NEOPRENE FLANGE GASKET (OR EQUIV)
32	160	COMM	HEX HEAD NUT, 3/4-10
31	152	COMM	HEX HD BOLT, 3/4-10 X 3.25 LONG
30	1	6666-3	SEN-DURE CLAMP ASSY (SHEET 2)
29	1	12818-1-7	SEN-DURE HEAT EXCHANGER (SHEET 2)
28	1	96008-024	COOLING FLUID TAP UNION
27	1	96008-023	SPECIAL 6" 150# "T" W/ PRESS TAP
26	1	96008-022	SPECIAL 6" 90° ELBOW/UNION
25	1	96008-04	ELECTRICAL ENCLOSURE ASS'Y (SHEET 2)
24	2	96008-019	ELBOW 6" 45° 150#
23	1	96008-018	UNION 6" FLANGED 150# -4
22	1	30732-HZ-24D	WORCESTER BALL VALVE ACTUATOR
21	1	8266T-150	WORCESTER BALL VALVE 6" FLANGE
20	2	STYLE 150-200	UNAFLEX SURA-FORM SINGLE ARCH EXPANSION JOINT 6"
19	2	AQ-310	ROTORK ACTUATOR NEMA 4, 24 VDC
18	2	STYLE 77-6	VICTAULIC FLEXIBLE COUPLING 6" SIZE
17	2	96008-017	UNION, VICTAULIC 6" SPECIAL
16	2	[illegible]	UNIFLEX 6" TEFLON EXPANSION JOINT
15	1	96008-016	T 6" FLANGED 150#
14	1	96008-015	UNION 6" FLANGED 150# VER #3
13	1	96008-014	UNION 6" FLANGED 150# VER 2(SHEET 2)
12	2	B5-511ACDA	SAUNDERS 6" BUTTERFLY VALVE
11	1	96008-013	UNION, 6" FLANGED 150# -1
10	2	96008-012	ELBOW, 90° 6" FLANGED 150# VER #1
9	1	604-C	GORMAN-RUPP PUMP
8	4	GB204-34	BARRY CONTROLS ENGINE MOUNT ISOLATOR
7	8	96008-011	BOLT, HOLD DOWN
6	4	96008-010	HOLD DOWN, BATTERY
5	1	PS2770	STROMAG PERIMAX FLEX COUPLING (NOTE2)
4	4	SEE NOTE 1	BATTERY GROUP 8D 12 VOLT 1200 TO 1300 AMP
3	1	96008-05	PIPE SUPPORT ASS'Y (SHEET 2)
2	1	[illegible]	CUMMINS 6 CYL DIESEL ENGINE
1	1	96008-03	PALLET ASSEMBLY

HOWLAND ASSOCIATES — PUMP/PALLET ASSY — AMBP — D — 96008-02 — REV A

Machinery Pallet Assembly

Inlet Side

Figure 3.2



Machinery Pallet Assembly

Outlet Side

Figure 3.3

During the course of Phase II, the Navy requested that attention be given to the feasibility of revising the design criteria to allow operation if the AMBP were located in a place where the tide cycle would periodically leave the AMBP grounded on the bottom without any seawater for cooling.

This requirement was investigated and it was determined that the fuel (or water) being pumped by the AMBP could be used as a secondary coolant instead of seawater if a means could be designed for this flow or a portion of the flow to be passed through a special heat exchanger designed for the purpose.

Accordingly, we worked with the engine heat exchanger manufacturer, SEN-DURE Products, Inc., to specify and design a special heat exchanger compatible with the engine and capable of using a small portion of the main fuel flow as the coolant.

Although SEN-DURE Products, Inc. designed the heat exchanger for this application as a special heat exchanger to be supplied with the Cummins engine, it was necessary to determine how the cooling fuel supply would be provided to the heat exchanger. The method selected uses a special jet pump concept implemented by a special 6-inch fitting in the main fuel flow line downstream of the main pump. This will be discussed in a separate section of this report.

(b) Design and Survival Current

During Phase I, the various operational sea states were defined but no sea current requirements were established. In order to calculate the drag on the hull and the various mooring line and hoseline forces acting on the AMBP, it is necessary to define the level of sea currents which the system must withstand.

For this purpose, the "design" sea current was established as 2.5 knots and the "survival" sea current as 5 knots. These are believed to be realistic current levels for the typical mooring location near the shoreline. Design current is defined as the current for which the system would be normally deployed and operated. Survival current, on the other hand, might require the deployment of unusual or extra anchoring capacity. However, assuming the location can be stabilized in the high current by the mooring system, the AMBP system would be expected to operate normally.

(c) Maintainability

During Phase I, it became clear that, if the hull were to meet the transportability requirement that the system fit into a standard ISO shipping container, there would be little room around the engine, pump, piping, and valves. Certainly, there would not be sufficient space for maintenance personnel to perform any significant maintenance tasks in the engine compartment.

Thus, at the outset of Phase II, a design criteria was adopted which required that the entire system, except for those components fixed to the hull of necessity, be capable of being easily disconnected from the hull and lifted out of the hull for servicing. This approach was followed in the design embodied in Figures 3.1 through 3.3.

The batteries are located on the machinery pallet with the engine so there are no high current electrical receptacles. There are electrical connectors for the supply and signals to the fuel level

transmitter mounted on the fuel tanks which are fixed to the hull, the bilge pump which must be located in the bilge off the pallet, and the strobe beacon which is mounted on the mast.

In addition, there is a quick disconnect in the fuel line which feeds the fuel tanks from the pallet, the engine exhaust pipe must be disconnected, and the main transverse pipes are fitted with Victaulic unions so that they may be easily disconnected from the hull fittings.

When the above electrical and fluid connectors have been disconnected, the whole equipment pallet can be lifted out of the hull by means of an overhead hoist or crane. The pallet is provided with an adjustable alignment system which mates with a track provided on the inner wall of the hull such that the pallet is guided back into alignment when it is reinserted into the hull compartment.

A small removable hatch and a hinged hatch are provided on the large main hatch to allow access to the engine oil dipstick and the filler points for adding engine oil or coolant. Two main circuit breakers for the main 12 VDC and 24 VDC feeds to the electrical enclosure are provided in a location that is accessible from the above hinged hatch. These breakers are used as master electrical switches.

Thus, periodic checks of the engine oil level and replenishment of oil can be accomplished without removing the pallet from the hull. Also, the main power to the electrical and electronic control system can be turned on and off from outside the hull, using the access hatch. After this power has been turned on, the system can be operated from outside the hull, using the shore-based terminal or an equivalent PC-based terminal equipped with a radio modem.

The other issue affecting maintainability is the fact that both the engine and the pump cannot be run for any significant period without pumping fuel or water through the main flow lines. Pumps of this type normally use seals that must not be allowed to run dry. In addition, the engine cooling approach adopted as discussed above requires the main pumped flow to supply the secondary cooling flow. Thus, testing of the unit or running for maintenance purposes will require that the pump be connected to a source of fuel or water and to produce sufficient pumped flow to supply a cooling flow to the heat exchanger.

4. MECHANICAL DESIGN CONSIDERATIONS

4.1 DESIGN RATIONALE

(a) Hull

During the preliminary design of Phase I, the conclusion was reached that the AMBP hull should be of welded construction. The question of whether steel or aluminum would be used was left open. As the layout and supporting analysis of Phase II progressed, it became clear that, with the relatively heavy equipment required inside the hull and its location, there was no real advantage to using aluminum for the bulk of the hull structure. In fact, for roll stability, it is necessary to provide considerable weight near the bottom of the hull, a task made easier by using fairly heavy steel structural elements for the bottom of the hull and the pallet on which the machinery is mounted.

The general boat-like shape, shown in the drawing of Figure 4.1, was adopted to minimize the drag produced by a current flowing toward one end or the other. Also, the upward sloping bottom in the bow areas helps prevent submarining or plowing under the water in a high current.

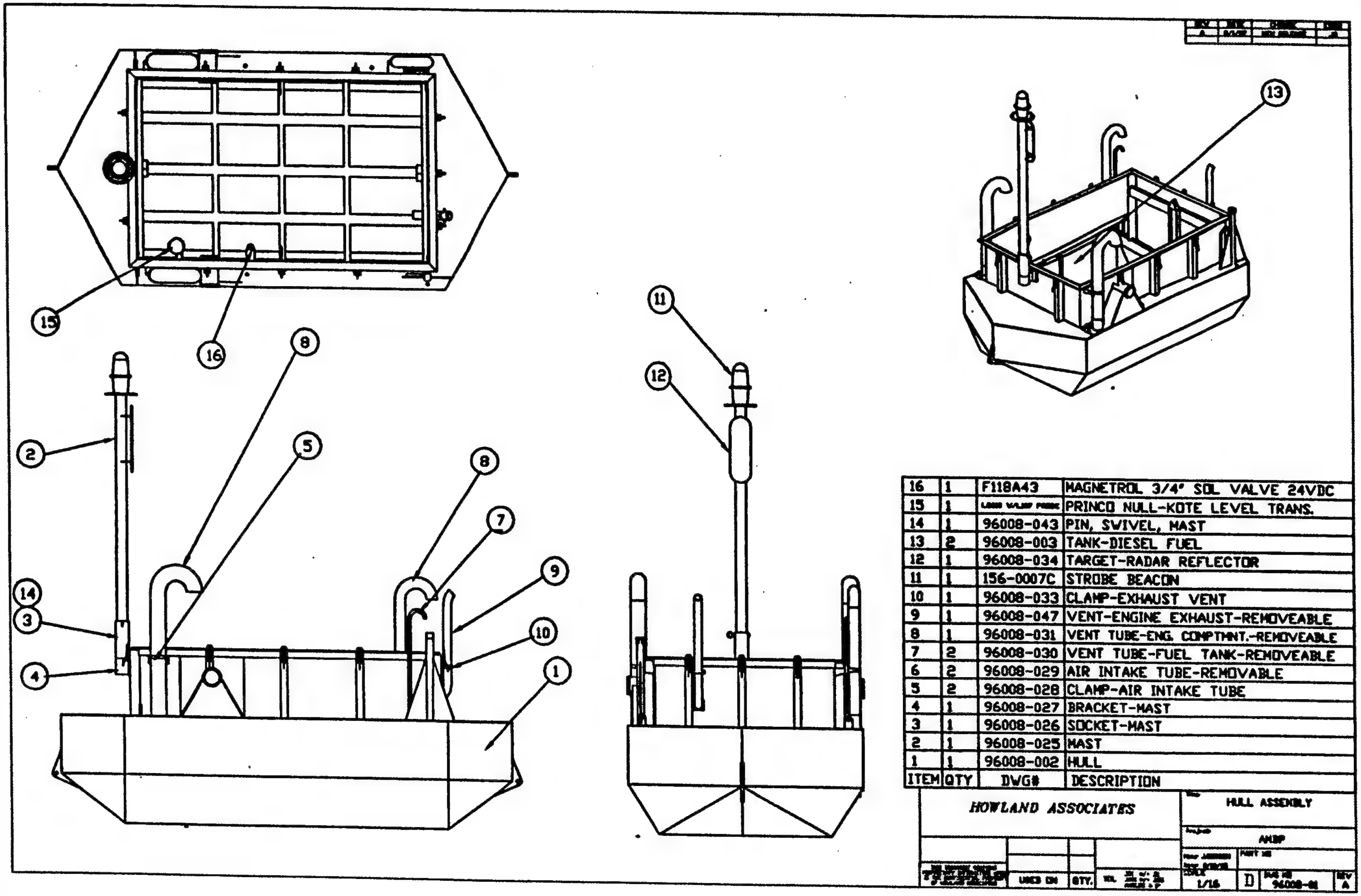
The bottom of the hull is made up of a lattice of 6-inch I-beams and angle. This lattice is covered by a 3/16-inch thick skin. The bottom structure is welded to corner plates or ribs and a rib plate at the mid-point along the length of the hull. 6-inch angles, also tied into the rib plates, form the deck structure. At the joints between the hull bottom, lower hull, and upper hull, where the two plates meet at an angle other that 90°, round tubing is used as the structural element to which the outer skin will be welded. This makes welding easier than would be the case if structural elements with a 90° corner were used.

Standard structural steel elements were used for the design where possible in order to allow the most general of fabricating capabilities to build the units. If the ultimate fabricator is equipped with sophisticated metal punching and forming equipment and is capable of forming special structural elements from steel plate, then it would be possible to redesign the hull and pallet structures to conform to the fabricating method. However, the strength of the new design should be verified. Also, any large weight variations should be checked for their effect on the floating stability of the whole system.

The bow structure is formed from 6-inch and 3-inch angle, a heavy 3/8-inch horizontal plate, and a vertical plate. The divider between the triangular center and side sections of the bottom is formed from round tubing.

The framework of the coaming above the deck is formed from 3-inch angle and covered by 1/4-inch plate. 3-inch channels are used along the side to support the toggle latches for the hatch cover.

The hose connection nipples, which must support the hoseline loads and transmit these loads to the coaming wall, are strengthened by means of a triangular box structure. The rear wall of this structure is formed by the coaming wall. The front plate and the two side plates are welded to the wall and deck.



ITEM	QTY	DWG#	DESCRIPTION
16	1	F118A43	MAGNETROL 3/4" SOL VALVE 24VDC
15	1	[illegible]	PRINCO NULL-KOTE LEVEL TRANS.
14	1	96008-043	PIN, SWIVEL, MAST
13	2	96008-003	TANK-DIESEL FUEL
12	1	96008-034	TARGET-RADAR REFLECTOR
11	1	156-0007C	STROBE BEACON
10	1	96008-033	CLAMP-EXHAUST VENT
9	1	96008-047	VENT-ENGINE EXHAUST-REMOVEABLE
8	1	96008-031	VENT TUBE-ENG. COMPTMNT.-REMOVEABLE
7	2	96008-030	VENT TUBE-FUEL TANK-REMOVEABLE
6	2	96008-029	AIR INTAKE TUBE-REMOVABLE
5	2	96008-028	CLAMP-AIR INTAKE TUBE
4	1	96008-027	BRACKET-MAST
3	1	96008-026	SOCKET-MAST
2	1	96008-025	MAST
1	1	96008-002	HULL

Hull Assembly

Figure 4.1

The lift points for the hull are located at the four corners and are tied into the angle iron framework near the corner rib plates. The mooring points are located at each end and are tied into the heavy vertical bow plates.

(b) Hatch Cover

The hatch cover is the only large component fabricated from aluminum. This was necessary in order to minimize the weight high on the structure to achieve roll stability. (See Section 6.) The cover, shown in Figure 4.2, is fabricated from 1/4-inch aluminum plate stiffened by a lattice of 3X2-inch formed channel to minimize deflections produced by local loading such as a man standing on the cover.

Two small hatches, located so as to allow the checking of engine oil level and the replenishment of engine fluids are located on the top of the hatch cover along with handholds.

Lift points are tied into corner gussets in the framework. Keeper brackets are provided for the toggle clamp hooks.

The hatch cover will be electrically isolated from the lip on the coaming by an elastomeric rubber gasket to prevent both leakage and galvanic corrosion. The steel latch blocks will be bolted to the hatch from the inside and will be electrically separated by an insulating pad.

(c) Machinery Pallet

The machinery pallet with all of the equipment mounted was shown in Figures 3.2 and 3.3. Figure 4.3 shows the pallet with the machinery removed. The heavy I-beam construction for the pallet provides two benefits. Firstly, the weight, near the bottom of the hull, aids in achieving roll stability. Secondly, when the pallet is lifted out of the hull for maintenance, the heavy equipment acts to produce bending of the pallet. The heavy beam construction stiffens the pallet, minimizing the deflection under this load.

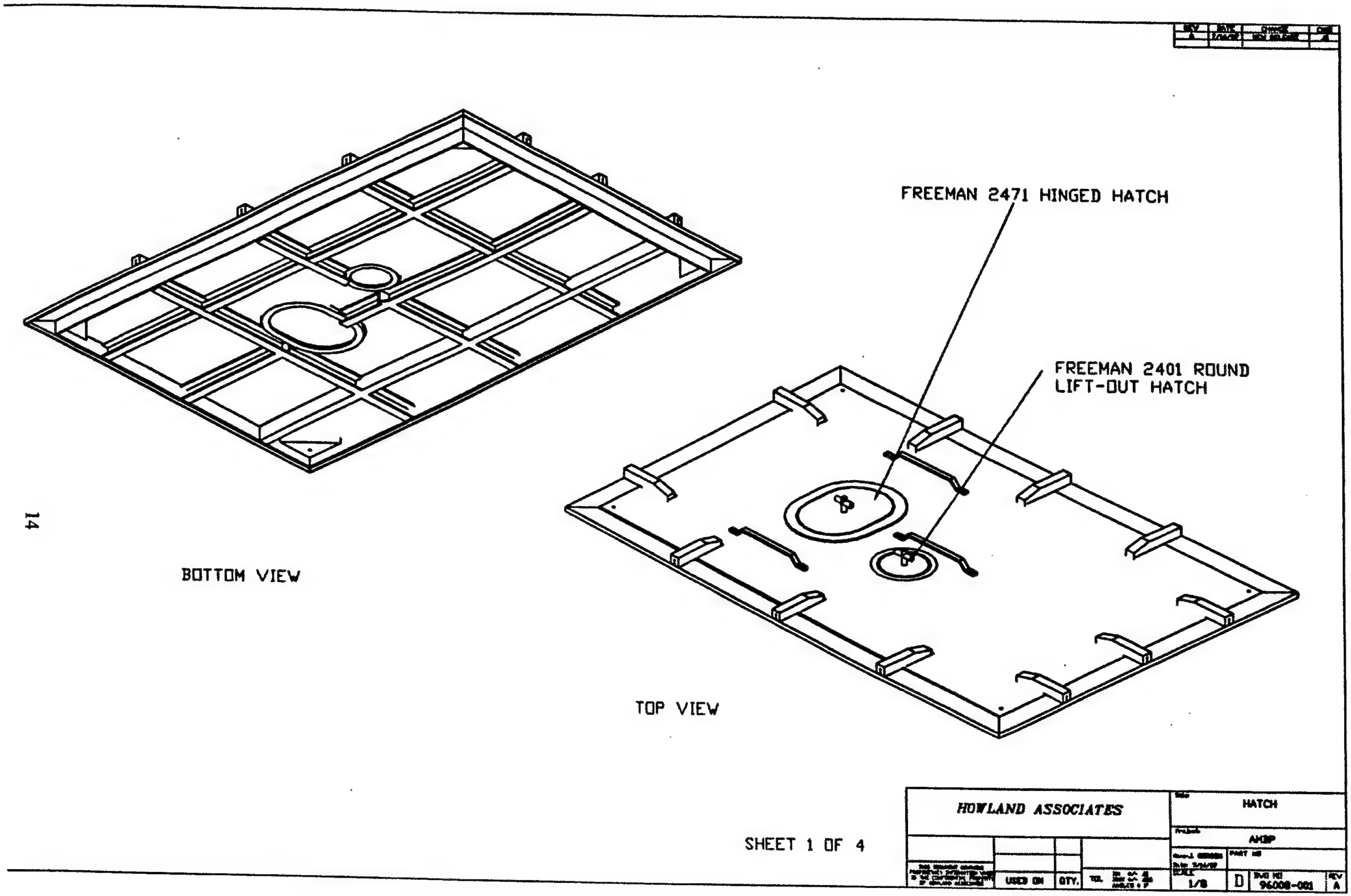
The lift points for the pallet are provided as extensions on the pallet end plates.

The principal mechanical design feature of the pallet design is the roller link provided at each end of the pallet to guide the pallet into place in the hull when it is re-inserted after maintenance. This roller link engages with a track provided on the inner wall of the hull. (See Figure 4.1) The alignment can be adjusted by means of the turnbuckles on the roller links such that the piping system lines up with the hoseline nipples welded on the hull.

Vertical and angular alignment is accomplished by means of leveling screws provided on each corner of the pallet.

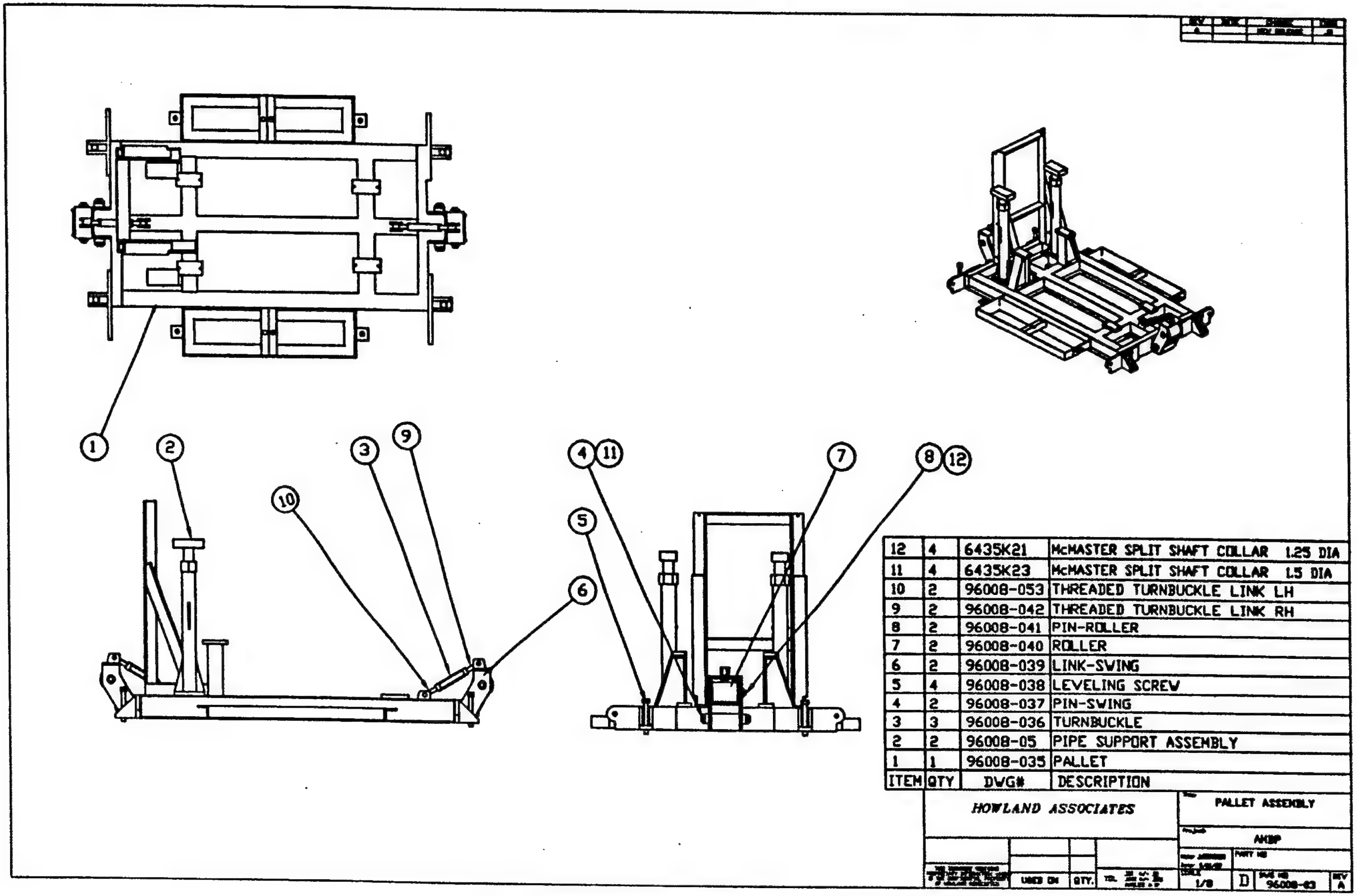
(d) Equipment Layout

The layout of the interior equipment on the pallet was affected by the need to meet a number of objectives. Firstly, it was necessary that the heavy engine and pump be located along the centerline of the hull. Otherwise, the imbalance would produce a list to one side which is not acceptable.



Hatch Cover

Figure 4.2



ITEM	QTY	DWG#	DESCRIPTION
12	4	6435K21	McMASTER SPLIT SHAFT COLLAR 1.25 DIA
11	4	6435K23	McMASTER SPLIT SHAFT COLLAR 1.5 DIA
10	2	96008-053	THREADED TURNBUCKLE LINK LH
9	2	96008-042	THREADED TURNBUCKLE LINK RH
8	2	96008-041	PIN-ROLLER
7	2	96008-040	ROLLER
6	2	96008-039	LINK-SWING
5	4	96008-038	LEVELING SCREW
4	2	96008-037	PIN-SWING
3	3	96008-036	TURNBUCKLE
2	2	96008-05	PIPE SUPPORT ASSEMBLY
1	1	96008-035	PALLET

Pallet Without Machinery

Figure 4.3

Secondly, when the engine and pump are mounted on the pallet, the locations of the pump inlet and outlet are prescribed. If the piping is allowed to extend beyond the end of the pump, then the overall length of the system and hull will be extended considerably. The alternative, which was selected for the layout shown in Figures 3.2 and 3.3, is to fold the piping back over the engine and pump. The penalty for this layout is a degree of crowding of the components.

The transverse layout was adopted for the by-pass piping with a full-bore ball valve provided as the by-pass valve. In this way, when the by-pass valve is open and the pump closed off by closing the two butterfly valves, a pig can be passed through the line providing that it can be passed through the hoseline fittings.

It should be mentioned that, during Phase I, the second butterfly valve in the outlet of the pump was proposed to be a hand operated valve instead of electrically actuated. During the early stages of this phase, this decision was questioned on the basis that it might prove desirable in some situations to be able to remotely isolate the pump from the pressurized main pumping line. However, it is unlikely that the remote operator would have any real indication of a slight leakage around the pump seal or other such conditions that would warrant this action.

In fact, the most likely reason for isolating the pump from the other piping would be for maintenance operations that require disassembly of the pump or replacement of the pump shaft seal. This will always occur when the equipment pallet has been removed from the hull and the butterfly valves are accessible for hand operation. Thus, the latest design requires that only one butterfly valve be electrically actuated. The wiring diagram and assembly drawings show the electrically actuated valve as an option.

The recommended method of vibration mounting the engine and pump is to connect the two components rigidly at the flywheel housing and to mount the combined assembly on vibration isolators. The engine and pump are connected at the flywheel by a flexible coupling which provides some torsional flexibility.

The piping system and valves are connected rigidly through flanged couplings to the pump and to each other. The inlet and outlet pipes are connected to the hoseline nipple, which is welded to the hull, by a victaulic union which is a split ring fitting that allows a significant clearance between the mating parts. This union is the fitting that must be disconnected in order to remove the pallet from the hull. In order to provide a degree of vibration isolation from the hull and a small amount of axial play for making up the victaulic fitting two sets of expansion joints are used in the piping. A single arch expansion joint is provided in the axial pipes just behind the butterfly valves. A teflon expansion joint is provided in the transverse lines just inside the two victaulic unions.

Because of the cantilevered nature of the piping system with its heavy valves and the presence of the flexible expansion joints, it was felt that the weight of the piping and valves needed to be supported in the area of the by-pass valve. It was felt necessary to use a system that would support the steady weight of the system but not restrict small motions due to vibrations. Since there is no structure overhead, to accomplish this support, it was decided to provide two adjustable pipe support pedestals as shown in Figure 4.3. These supports are adjustable in length and do not clamp or restrain the piping in any way. It would probably be desirable to cover the top of the support with a hard rubber pad to eliminate any chatter due to vibration. In operation, the pipe

supports would be adjusted in length until they just begin to support the weight of the piping and valves.

4.2 STRENGTH

It is clear to the practiced mechanical designer that the structure of the hull and the equipment pallet is generally overdesigned from a strength standpoint. However, as has been discussed, the excess weight near the bottom of the hull was an advantage. Because the hull is overdesigned, it is possible to analyze the major stresses by using very conservative idealized models as summarized in this section.

(a) Hull

The bottom of the hull is formed by a lattice structure formed with three main lengthwise I-beams. These are standard W 6X12 I-beams. When floating, the buoyant forces which take the form of a distributed pressure acting on the bottom of the hull support the weight of the hull and its contents as well as any mooring forces which act downwardly on the hull structure.

By neglecting the strengthening provided by the surrounding structure, the strength of the hull I-beams can be idealized by considering them to have fixed ends and loaded with a distributed force equal to 1/3 of the total downward force on the hull. In a 5 knot current, the total load has been calculated to be 12,790 lbs. including the mooring force. (See Section 7)

The maximum bending stress in a fixed-fixed beam subject to distributed loading can be expressed by the relationship (See Case 33 of Roark (8))

$$\sigma = Mc/I = 5353 \text{ psi} \tag{4.1}$$

where

$$M = 1/12\, W\ell \tag{4.2}$$

$W = 4263$ lbs., 1/3 of the total force
$\ell = 109$ inches
$c = 3$ inches (for a 6-inch beam)
$I = 21.7$ in.^{4} (Ref. (9))

This stress level is much lower than the yield stress for AISI 1020 steel of 30,000 psi (10) which we will consider to be the allowable stress level.

The effect of dynamic loading due to wave motion is also of interest. This, of course, very difficult to analyze accurately since the wave characteristics are complex and the response of the AMBP to the complex wave motion is difficult to analyze. In discussions of the dynamic response of moored buoys to wave motion, the concept of virtual mass which is meant to represent a volume of entrained water which accelerates up and down with the buoy is introduced. (2) If the downward motion of the hull accelerates the fluid under the hull, then it is possible to get an increased pressure between the water and the hull above the hydrostatic pressure due to the unsteady or acceleration term.

Suppose the hull were floating in a sea state of 5 and accelerating up and down sinusoidally, following a typical average of the 10% largest waves for this sea state. According to Ref.(2), the amplitude of these waves is 16 ft. and the most stringent or shortest period is 3.7 secs. which

corresponds to an angular frequency of 1.69 rad./sec. If the hull follows these waves, the amplitude of the sinusoidal acceleration can be expressed as

$$a = \omega^2 A = 46 \text{ ft./sec.}^2 \tag{4.3}$$

where $\omega = 1.69$ rad./sec.
$A = 16$ ft.

This corresponds to about 1.4 g's which would have to be produced by increased pressure on the hull bottom during the portion of the cycle when the acceleration is upward. Thus, we could rationalize a total force, static plus dynamic, of 2.4 g's in this case which would correspond to a bending stress of about 12,847 psi., still safe. During the downward acceleration portion of the cycle, of course, the pressure would have to be negative which is not possible. Thus, the hull would break free of the water during this portion of the cycle and the response, in actuality, would be quite complex.

Another unusual case which should be examined is the case where the tide goes out and leaves the hull stranded on a rock such that a concentrated load is produced on the bottom of the hull. For the case where the beam is still considered as a fixed-fixed beam but is loaded by a concentrated load equal to the entire weight of the hull and contents, 11,390 lbs., Eq.(4.1) can still be used where

$$M = 1/8\, W\ell \tag{4.4}$$

and $W = 11{,}390$ lbs.

Substitution into Eq.(4.1) gives a bending stress of 21,454 psi. which is still below the yield stress.

A local stress of interest is the shear stress in the lifting eyes when the entire hull and contents are lifted with a crane. The lifting eyes have a 1-inch dia. hole centered on a 3-inch tab which is 1-inches thick. Thus, the shear area of a pin in the 1-inch hole is approximately

$$A_s \cong (2)(1)(1) = 2 \text{ in.}^2$$

Since the load, P, for each of these eyes is 1/4 of the total weight or 2848 lbs., the shear stress becomes

$$\tau = P/A_s = 2848/2 = 1424 \text{ psi.} \tag{4.5}$$

This should be compared with a limit of about half the yield stress or 15,000 psi and is quite safe.

Another local shear stress of interest is that produced at the mooring eye. In this case, the mooring eye also has a 1-inch dia. hole located approximately 1.39 in. back from the nose of a 1-inch thick tab. Thus, the shear area is approximately

$$A_s = (2)(1)(0.89) = 1.78 \text{ in.}^2$$

The worst mooring load occurs at a survival current of 5 knots and is 5000 lbs. (See Section 7)

Thus, Eq. (4.5) can be used to get the shear stress

$$\tau = 5000/1.78 = 2809 \text{ psi.}$$

which is quite safe.

(b) Hoseline Connection

The hoseline Connector fittings pass through the wall of the coaming. They are supported by a triangular box structure as shown in Dwgs. 96008-051 and 96008-052. The entire box structure is welded together and to the deck and coaming wall which forms the rear wall of the box.

The hoselines are loaded by drag from the current. In the worst case of a 5 knot survival current, the horizontal component of hoseline tension, P, has been shown to be 3160 lbs. (See Section 7) The distance from the hoseline fitting to the wall of the hull coaming is about 12 inches. Thus, the above force produces a moment on the structure of

$$M = 3160(12) = 37920 \text{ in-lbs.} \qquad (4.6)$$

This moment is reacted by tension and compression stresses between the side plates and the coaming wall. If we assume that the effective moment arm for the reaction of this moment occurs at the mean separation between the two side plates or

$$a = (24.25 + 6.5)/\,2 = 15.375 \text{ in.}$$

then the reaction forces on the wall can be approximated as

$$F = M/a = 2466 \text{ lbs.} \qquad (4.7)$$

The normal area for each of these reaction forces is

$$A \cong (0.5)(15.26)/\cos 30 = 8.8 \text{ in}^2 \qquad (4.8)$$

Thus, the average tensile and compressive stress between the side plates and the coaming wall at survival current is

$$\sigma = F/A = 280 \text{ psi.} \qquad (4.9)$$

This stress is actually calculated using a very conservative model since the welded joint between the fitting and the coaming will also react some of this moment. Thus, the fact that this stress level is negligible suggests that the various reaction stresses on the hoseline fitting structure will be negligible.

(c) Mast

Another area of concern for strength is the mast and its fittings, shown in Figure 4.1 and described in detail in Dwgs. 96008-025, -026, and -027.

The weight of the mast can be found as

$$W = \gamma \pi Dt L = 57 \text{ lbs.} \tag{4.10}$$

where $\gamma = 0.3$ lbs./in.^{3} for steel
$D = 4$ in.
$L = 97$ in.
$t = 0.156$ in.

If we assume that the AMBP might experience 3 g's sideways acceleration due to rough seas, then the distributed sideways load would be three times the weight or 171 lbs. A 3g acceleration is felt to be quite conservative.

In addition to the inertial force, a drag force from a high wind is also possible. This force can be expressed as

$$D = 1/2\ \rho ACV^2 = 65.5 \text{ lbs.} \tag{4.11}$$

where $\rho = 0.0735$ lb./ft^3 for air
$A = 2.69$ ft^2
$C = 1$, conservative for cylinder
$V = 146$ ft/sec, conservative 100 MPH wind

Adding these two very conservative distributed side loads together gives

$$F_t = 171 + 65.5 = 236.5 \text{ lbs.}$$

This is the shear force at the base of the mast. Since the mast acts as a cantilever beam, Case 3 of Roark (8) can be used to find the maximum bending moment at the base of the mast

$$M = 1/2\ F_t L = 11{,}470 \text{ in-lbs.} \tag{4.12}$$

and the corresponding bending moment is given by Eq. (4.1) as

$$\sigma = Mc/I = (11{,}470)(2)/(3.48) = 6592 \text{ psi}$$

since $I = 1/4\ \pi\ (R_o^4 - R_i^4) = 3.48$ in.^{4}

which is quite low in comparison with a typical yield stress for 1020 steel of 30,000 psi.(10)

The mast is attached to the wall of the hull coaming by a 1-inch fillet weld around the periphery of the foot of the mast bracket. (Dwg. 96008-027) The length of this weld is approximately 26 inches giving a shear area of approximately 26 in^2. The shear stress under the shear load of 157 lbs. is, thus, only 6 psi. which is negligible.

(d) Hatch Cover

The hatch cover, shown in Figure 4.2, is a 1/4-inch aluminum plate with an angle iron frame around the edge and a lattice of stiffeners made from formed channel. Neglecting the stiffeners conservatively, the cover can be analyzed as a rectangular plate with supported edges. The weight of the cover is 372 lbs. For a rectangular plate with dimensions, a x b, the maximum bending stress under a distributed load, w, is given by Case 36 of Roark (8) as

$$\sigma = \beta\, w\, b^2 / t^2 = 1882 \text{ psi.} \qquad (4.13)$$

where

$a = 117$ in.
$b = 76$ in.
$\beta = 0.485$ for $a/b = 1.54$ from Roark
$w = W/ab = 372/8892 = .042 \text{ lb/in}^2$ (4.14)

The bending stress, given by Eq. (4.13), is, of course, quite low compared with the yield stress for optimum aluminum alloys such as 5154-0 (17,000 psi.) or 5456-0 (23,000 psi.) (11)

Another loading of interest is that due to a man standing in the center of the hatch cover. Assuming a 200 lb. man standing with his load distributed over a 10-inch diameter footprint, Case 37 of Roark gives the bending stress

$$\sigma = 3W/2\pi m t^2 \,[\,(m+1) \ln (2b/\pi r_o) + (1 - \beta m)] = 4715 \text{ psi.} \qquad (4.15)$$

where

$W = 200$ lbs.
$m = 1/\nu = 3.33$, reciprocal of Poisson's ratio
$t = 0.25$ in.
$r_o = 5$ in.
$\beta = 0.168$ for this case from Roark

Of course, the bending stresses of Eqs. (4.13) and (4.15) are in the same direction and occur at the same place, the center of the cover. Thus, they should be added to give a resultant bending stress of 6597 psi. This stress level is safe compared with the yield stresses cited above. No permanent deformation or denting should occur so long as the weight is distributed over a reasonable footprint.

For the hatch cover, it is also interesting to determine the deflection under these loads. This is also maximum at the center if a man is standing in the center. Without going through the details of the calculations, the two cases cited above from Roark (8) can also be used to calculate the center deflection. Superimposing the results for the two cases gives a center deflection of 1.97 inches. This is not excessive, of itself, but it should be remembered that the stiffening effect of the channel members welded to the inside of the cover plate was neglected. This should reduce the deflection considerably.

(e) Pallet

In operation, the pallet is supported by the hull bottom and strength is not a real concern. However, when the pallet is lifted by the end bars in order to remove the machinery for maintenance, the weight of the pallet and the machinery mounted on it, 3590 lbs., will produce a bending stress on the pallet beams. The critical dimensions for the pallet are given in Dwg. 96008-035.

The bending stress level can be predicted conservatively by taking 1/3 of the entire load or 1197 lbs.on each of the main beams which are assumed to be simply supported. Most of the load acts at the two cross-members so that half of the applied load or 598.5 lbs. is assumed to act at each of these. Case 12 of Roark (8) can be used to determine the maximum bending moment which occurs under each load by superimposing the results from the two loads. The result is

$$M_{max} = Pab/L + P(a/L)(L-a) = 15,750 \text{ in-lbs.} \qquad (4.16)$$

where
$P = W/2 = 598.5$ lbs.
$a = 16.6$ in.
$b = 62.6$ in.
$L = 78.6$ in.

The corresponding bending stress at these points is

$$\sigma = M_{max}\, c/I = 2177 \text{ psi.} \qquad (4.17)$$

where the distance from the neutral axis,c, and the moment of inertia,I, for the W 6X12 I-beam are 3 inches and 21.7 in^4 as before. This stress level is, of course, no problem.

Another area of potential concern is the shear and bending stresses on the battery nests which are welded to the side of the pallet. For one nest, the shear area is approximately (See Dwg. 96008-035, Sheet 4)

$$A_s = (4)(3)(0.375) = 4.5 \text{ in}^2 \qquad (4.18)$$

Since the weight of two batteries, W, is approximately 260 lbs, the shear stress in the welds can be calculated as

$$\tau = W/A_s = 57.7 \text{ psi.} \qquad (4.19)$$

which, of course, is no problem. Because the weight is cantilevered, there is also a bending stress on the weld. Assuming that it tries to rotate about the bottom point of the angle member and neglecting the resistance of the vertical portion of the angle, conservatively, the resulting tensile force on the horizontal portion of the angle members becomes,

$$F = 260(5.5)/3 = 477 \text{ lbs.}$$

Since the area of the horizontal portion of the angle member is about 2.25 in^2, the tensile stress becomes

$$\sigma = 477 / 2.25 = 212 \text{ psi.} \tag{4.20}$$

which is again not a problem.

4.3 THERMAL CONSIDERATIONS

The specified environmental air temperature range for the AMBP is -30F to 120F. The electronic components which, in general, are capable of operating in a range between 32F and 160F could be a problem on both ends of the environmental range.

For the low end, of course, they simply will not operate if the electronics cabinet reaches temperatures close to the lower limit of -30F. Thus, it is necessary to heat the electrical cabinet whenever the AMBP is operational under conditions where the ambient air temperature is below this temperature. Otherwise, the system could not be turned on when it has not been running for a period of time. Once the system has been on for a period of time the heat generated by the electronics units themselves may be capable of keeping the units above the lower operating limit. However, a thermostatically controlled heater will ensure that, whenever the master switch is on the cabinet will be kept within the operating range.

The size of the required cabinet heater can be determined from the heat transfer conditions on the cabinet. The electronics cabinet is approximately 4 ft. W x 4ft. H x 1 ft. D. Thus, the total vertical wall area, A, of the cabinet is 40 ft^2.

The heat lost from a heated cabinet at a temperature, T_1, to the surrounding ambient air a temperature, T_2, is given by

$$q = U A \Delta T \tag{4.21}$$

where
U = overall heat transfer coefficient, Btu/hr.ft^2F
A = vertical area of the wall, 40 ft^2
ΔT = temperature difference across the wall = $T_1 - T_2$

The overall heat transfer coefficient can be found from the relationship

$$\frac{1}{U} = \frac{1}{h_{in}} + \frac{1}{h_{out}} + R \tag{4.22}$$

where
R = thermal resistance of the cabinet walls
h_i = convective heat transfer coefficient of the walls

If we assume that the cabinet walls are insulated, the thermal resistance, R, is given by

$$R = b / k = 8 \tag{4.23}$$

where
b = insulation thickness, ft. = 0.2 ft. as a typical practical value

k = conductivity of insulation =0.025 Btu/hr.ft.F, typically

Realizing that a typical natural convection heat transfer coefficient on a vertical wall is approximately 1 Btu/hr.ft$_2$F, Eqs. (4.22) and (4.23) can be used to get

$$U = 0.1 \text{ Btu/hr.ft}_2\text{F} \quad (4.24)$$

Substituting into Eq. (4.21) then gives the heat loss to the ambient,

$$q = (0.1)(40)(70) = 280 \text{ Btu/hr.} = 82 \text{ watts} \quad (4.25)$$

If we assume that the ambient temperature is -30F and we wish to hold the interior of the electronics cabinet at +40F. This analysis was used to size the cabinet heater which was selected to be a 100 watt unit. The insulation could be practically any type of architectural or industrial insulation. It might be convenient to cut appropriate pieces and affix them to the cabinet surfaces with Velcro patches so that the insulation could easily be removed and used only for cold weather operations.

At the high end of the environmental range, the air temperature is not high enough by itself to be a problem. However, the system is enclosed in a compartment with a large area exposed to solar radiation which could theoretically raise the compartment temperature to levels above the 160F limit. Thus, we have analyzed a worst case limiting scenario to determine whether forced cooling will be necessary.

The first case examined is where the diesel engine is not running. The highest possible solar radiation heat input will occur when the sun is directly overhead at the equator on the day of the year when the sun is closest to the earth. (Assuming that these conditions can all occur at once!) This would produce a total radiation flux of about 329 BTU/hr.ft^2 (14) Assuming that the emissivity of the hatch cover is a typical 0.85, the radiation input to the AMBP would be approximately 280 BTU/hr.ft^2.

In addition to the extreme solar radiation conditions, the following limiting conditions can be assumed:

> Air temperature, T_a, is taken to be 120F, the highest specified.
> Water temperature, T_w, is taken to be 90F. a very high value for ocean water.
> There is assumed to be no wind, a conservative assumption.

The overall heat balance for the AMBP is shown in the schematic diagram of Figure 4.4. where the various heat fluxes, all defined in units of BTU/hr.ft^2 and normalized per unit area of the <u>deck,</u> are defined as follows: (15)

q_{rad} = heat radiated from the cover to the sky

$$= 0.17 \in [(T_c/100)^4 - (T_{sky} / 100)^4] \quad (4.26)$$

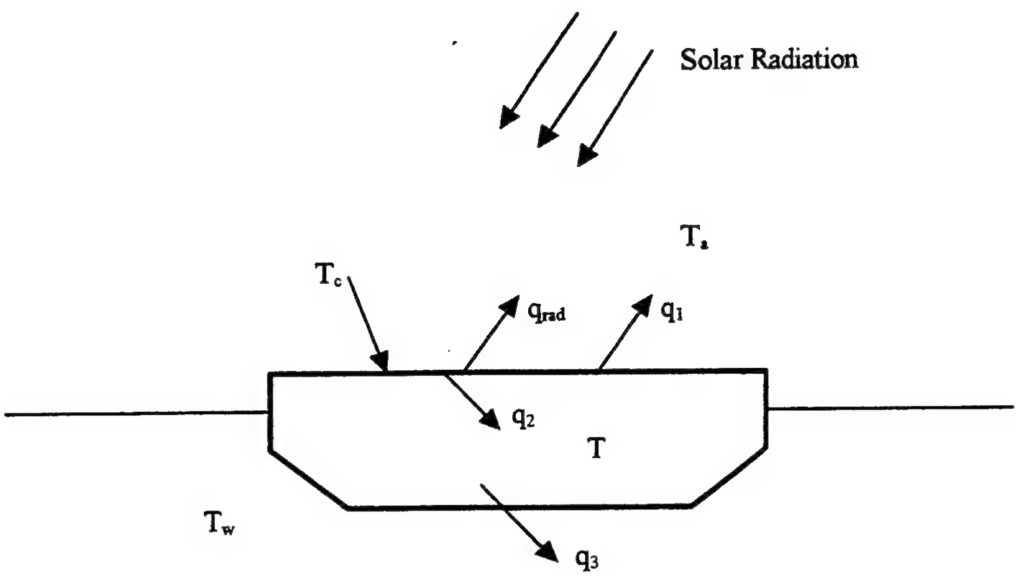
where $\in$ = emissivity of the cover = 0.85
T_c = temperature of the cover , expressed as $F_{abs.}$ or Rankine for this equation.
T_{sky} = temperature of the sky, usually taken at about -100F or 360R.

q_1 = heat convected from the cover to the atmosphere

$$= h_1 (T_c - T_a) \tag{4.27}$$

q_2 = heat convected from the cover to the interior of the hull

$$= h_2 (T - T_w) \tag{4.28}$$



Schematic Diagram Showing the Heat Transfer
on the Hull
Figure 4.4

q_3 = heat convected from the vessel interior to the walls in contact with the ocean water or,

$$q_3 = h_3\, r\ (T - T_w) \qquad (4.29)$$

The heat transfer coefficients in Eqs. (4.27) through (4.29) can be approximated as follows: (15)

$$h_1 = 0.38\ (T_c - T_a)^{0.25} \quad \text{for a horizontal surface facing upward.} \qquad (4.30)$$

$$h_2 = 0.2\ (T_c - T)^{0.25} \quad \text{for a horizontal surface facing downward.} \qquad (4.31)$$

$$h_3 = 0.25\ (T - T_w)^{0.25} \quad \text{for a mixture of horizontal and vertical surfaces.} \qquad (4.32)$$

In Eq.(4.29), the ratio, r, represents the area ratio between the deck and the portion of the hull actually in the water. This can be derived from the geometry of the hull and the waterline location determined in Section 6. of this report to be approximately 1.67.

A heat balance on the cover requires the condition,

$$q_{rad} + q_1 + q_2 = 329 \ \text{Btu/hr.ft}^2 \qquad (4.33)$$

A heat balance on the hull interior requires that

$$q_2 = q_3 \qquad (4.34)$$

Because of the exponents in the radiation and convection coefficient relationships, it is difficult to solve these equations explicitly. However, it is relatively easy to assume some temperature values and rapidly iterate the equations to determine a set of conditions that satisfies the relationships. If one does this for the above assumed conditions, the resulting cover and interior temperatures can be found to be:

$$T_c \cong 170\text{F}$$
$$T \cong 119\text{F}$$

Thus, the simplest extreme radiation condition suggests that there should be no problem on the interior of the AMBP. The interior temperature will be slightly below the air temperature. The surface of the cover will become fairly hot under these conditions. However, that would be improved, if desired by painting with a reflective coating to increase its reflectivity so that less incident solar radiation is absorbed.

The largest effect on the system when the engine is running is that approximately 1000 lb./hr. of air is pulled through the compartment. Even if the temperature of this air only increased by 10F over the ambient temperature, it would absorb about 2400 Btu/hr. which is more than the entire heat input from the deck to the interior compartment under the extreme conditions examined above.

Most of the excess engine heat is carried away in the exhaust gases and the coolant fluid, which, in this case, is the pumped flow passing through the AMBP. Since, the air intake system has been sized in accordance with Cummins recommedations, we expect that the engine heat should not be a

problem. The large number of metal surfaces within the compartment which are kept at ocean temperature or exterior air temperature by forced convection should maintain the temperature at safe levels.

Since it would be difficult to provide considerably more ventilation area without interfering with the hatch cover and other structures on the deck, further cooling of the interior compartment would require forced ventilation and, perhaps, baffling to ensure that the air passed efficiently over the required surfaces. This could only be a problem when the ambient temperature is well over 100F. Thus, we would recommend that the prototype AMBP unit be tested under warm weather conditions by monitoring the interior temperatures to enable prediction of the temperatures which will result under the most extreme running conditions by extrapolation. For example, it should be possible to analyze the test conditions to determine the heat load imposed by the engine. This heat load, which is not available otherwise, could then be used to predict the compartment temperatures under the most extreme conditions.

4.4 ENGINE COOLING

As discussed in Section 3.2, the normal method of cooling a marine diesel engine is by means of a keel cooler which is a heat exchanger on the bottom of the hull or a sea-water circulation system which supplies sea-water to a heat exchanger on the engine. Because our "vessel" is moored at a stationary location, the original concept was to use a forced seawater circulation system. However, when the design criteria was changed to require operation when left high and dry by a receding tide, it was clear that sea-water could no longer be used as the secondary coolant. Instead, it was decided to use the pumped fluid as the secondary coolant.

In cooperation with the heat exchanger manufacturer which supplies the OEM heat exchangers to Cummins, SEN-DURE Products, Inc., we have developed a cooling system that makes use of a small fraction of the pumped flow which is bled off the main flow by means of a special fitting. This cooling flow is passed through a special SEN-DURE heat exchanger. Because this heat exchanger is larger that those normally used on forced sea-water systems, it is mounted on the equipment pallet alongside the engine. The engine is then ordered in the keel cooled version (no sea water pump) but with the special heat exchanger instead of the normal keel cooler.

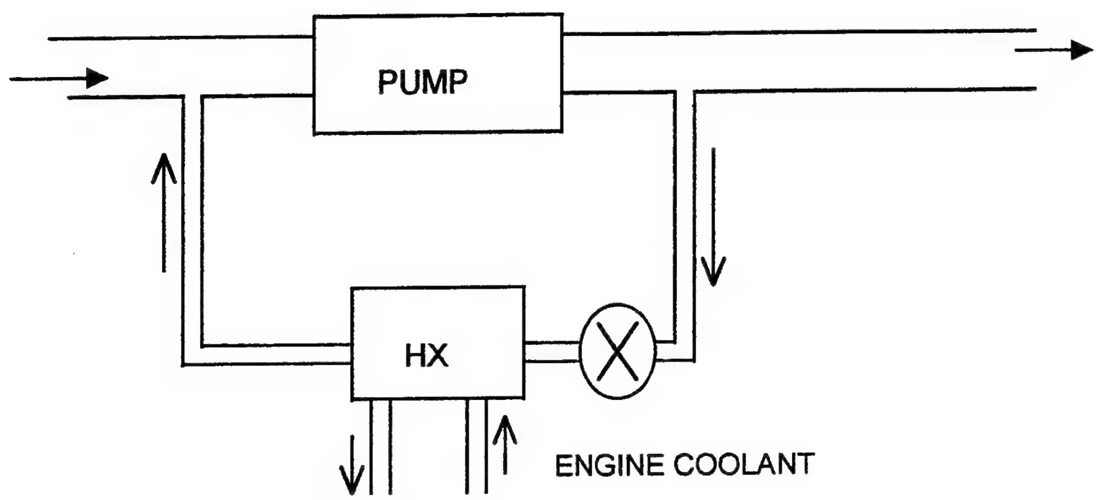
At the outset of this task, it was assumed that the simplest method was to simply pass the entire pumped flow through a heat exchanger designed to cool the primary engine coolant pumped from the cooling jacket. This would require a 600 GPM heat exchanger mounted in the suction or discharge line of the pump. This heat exchanger was sized and found to be so large that would require lengthening of the entire piping layout significantly.

Moreover, it was found that the required heat transfer does not require such a large heat exchanger. Thus, it was decided to use only a portion of the pumped flow for cooling purposes. There are two basic methods for diverting a portion of the flow, shown schematically in Figure 4.5.

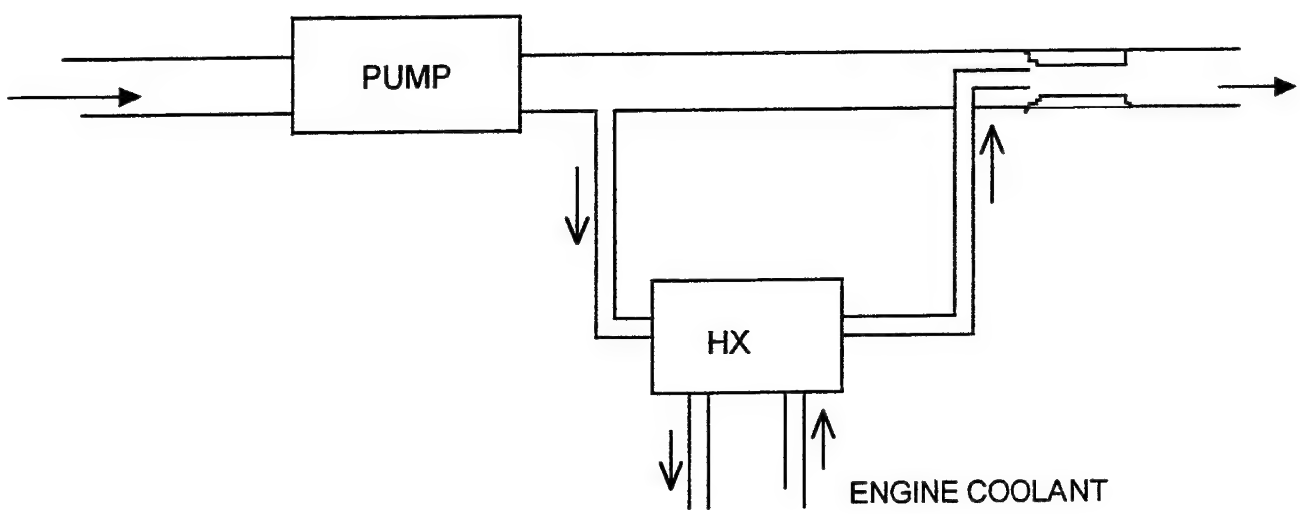
(a) By-passing where a small portion of the main flow is tapped at the pump discharge, passed through the heat exchanger, and returned to the line at the pump suction, shown in Figure 4.5(a).

(b) In-line bleeding where a small portion of the main flow is tapped upstream of a restriction, passed through the heat exchanger, and returned to the line at the restriction where the pressure is lower. This is shown in Figure 4.5(b).

Method (a) is not desirable in our case since the by-passed portion or percentage of the flow simply circulates around the pump, never exiting the system. Thus, the net output of the AMBP would drop by this percentage. Since the selected pump operates at the specified 600 GPM for the design pressure conditions, it would no longer meet the specification. Moreover, we do not require the very large pressure drop which could be produced by this method to drive the fluid through the



(a) By-Pass Configuration



(b) In-Line Restriction Configuration

Coolant Flow Configurations
Figure 4.5

heat exchanger. Thus, the excess pressure would have to be throttled with a valve in the by-pass line leading to a low energy efficiency.

Method (b), on the other hand can be impemented without any net change in the pumped flow and by introducing only a very small added pressure drop to the pumped line. Thus, we elected to proceed with this concept.

The energy balance for the heat exchanger can be written:

$$q = w\,c\,\Delta T = f\,W\,c\,\Delta T \qquad (4.35)$$

where,

q = maximum engine cooling requirement

w = mass flow rate of pumped liquid diverted to the heat exchanger, lbm/hr.

W = mass flow rate through pump = $\rho\,Q$, lbm/hr.

$f = w/W$ ratio of diverted flow to pump flow

c = specific heat

Q = volume flow through pump

ΔT = temperature rise of diverted fluid across heat exchanger

ρ = density of the pumped fluid

Thus

$$\Delta T = \frac{q}{\rho Q c f} \qquad (4.36)$$

Using the following values yields the results of Table 4.1, for several realistic values of flow fraction, f:

q = 270,000 BTU/hr

ρ = 50 lbm/ft^3

Q = 600 GPM = 80.2 ft /min

c = .5 BTU/lbm F

TABLE 4.1

TEMPERATURE RISE OF THE DIVERTED FLUID AS A FUNCTION OF MASS FLOW RATIO

f	ΔT
0.15	15.0°
0.10	22.4°
0.06	37.4°

The resulting temperature rises for the diverted fluid are quite acceptable. The resulting temperature rise for the overall flow when the diverted flow is mixed back into the main flow will, of course, be much smaller.

The nominal heat exchanger flow was selected to be 10% of the overall pumped flow. In cooperation with SEN-DURE, the supplier of heat exchangers for the Cummins marine engines, an appropriate heat exchanger was developed to meet the required cooling load. A drawing of this heat exchanger is shown in Figure 4.6. Its major characterisitics are as follows:

Nominal diameter of heat exchanger = 6 inches

Length = 31 inches

Number of tubes = 264

Diameter of tubes = .25 inches

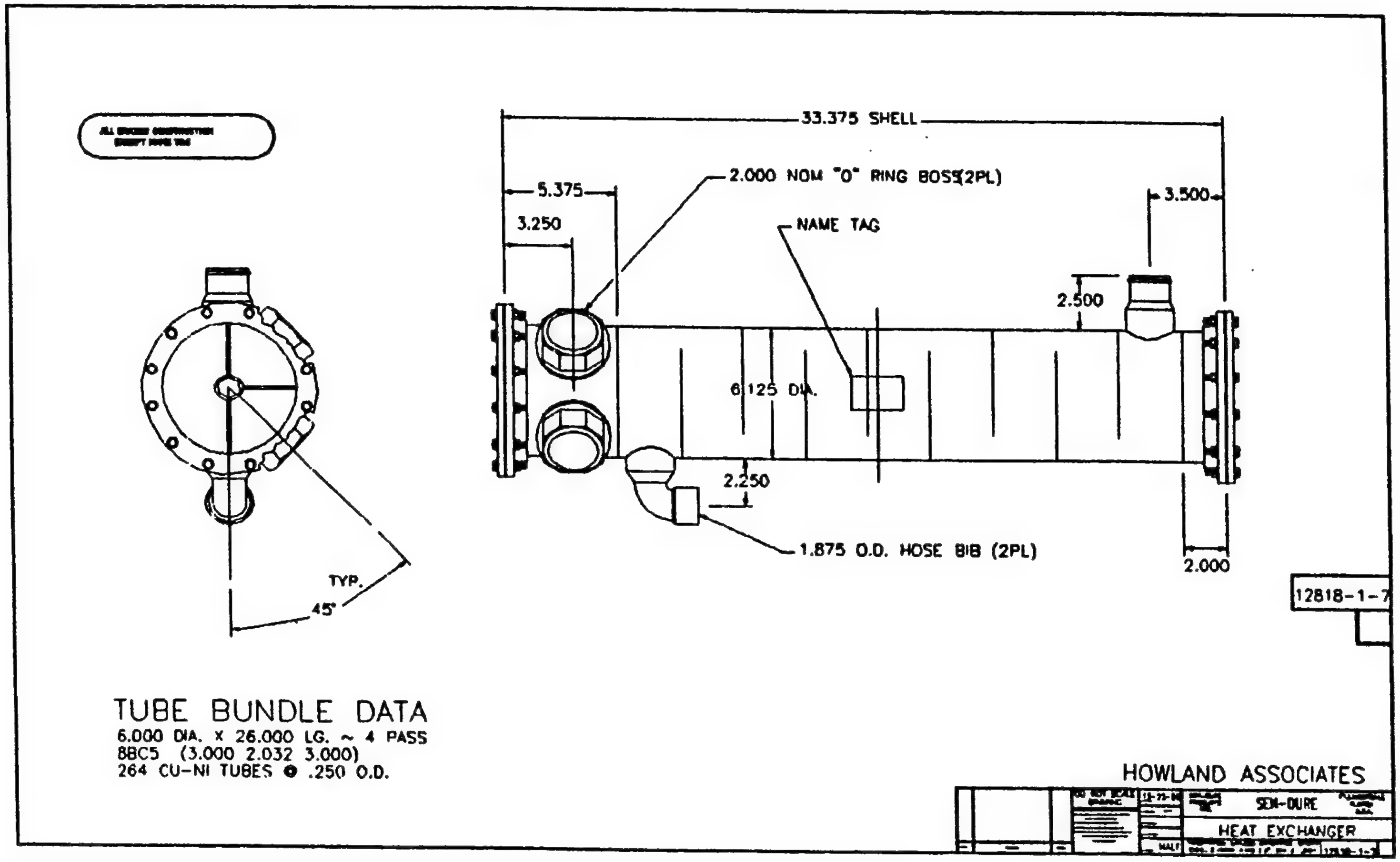
Tube material - Cu - Ni

At a flowrate of 60 GPM, the heat exchanger requires a pressure drop of about 2 psi to drive the flow through it. Thus, the special fitting in the AMBP flow loop required to supply the heat exchanger must be capable of developing this pressure difference. This special fitting, shown in Figure 4.4(b) operates on a principle similar to a jet pump. The device is shown schematically in greater detail in Figure 4.7.

The area restriction at A_4 results in a velocity increase and a pressure decrease relative to the pressure, P_1, slightly upstream of the jet insertion point. Thus, a pressure difference is created between P_1 and P_2 which forces the tapped fluid through the heat exchanger. The reentering fluid simply mixes with the main flow and is fully mixed at A_5, slightly downstream from the restriction.

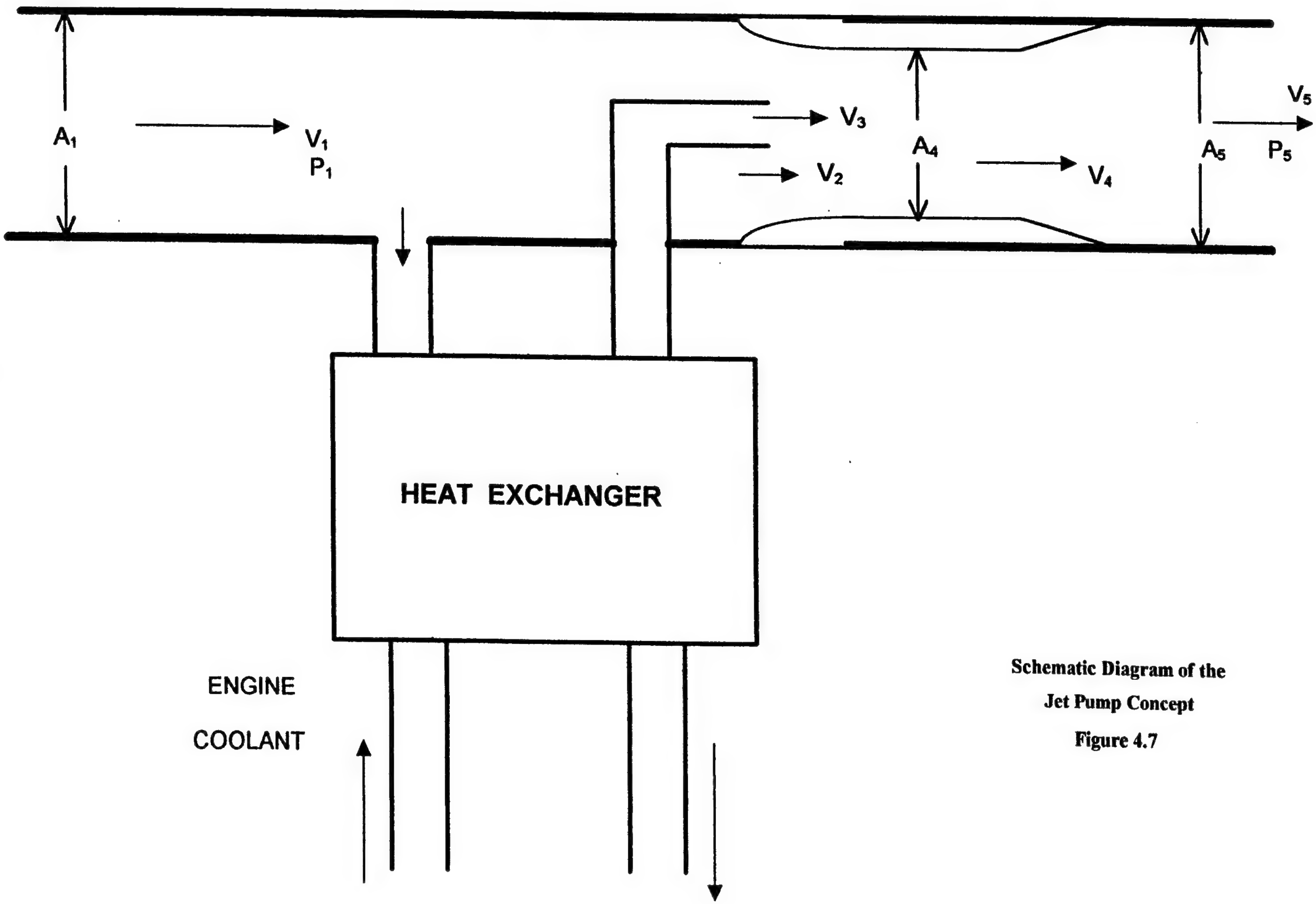
Bernoulli's equation can be written between point 1 and point 2 as follows:

$$P_1 - P_2 = \rho/2\,(V_2^2 - V_1^2) \qquad (4.37)$$



SEN-DURE Heat Exchanger

Figure 4.6



Schematic Diagram of the
Jet Pump Concept
Figure 4.7

For $Q_1 = 600$ GPM or $1.337\ ft^3/sec.$, and a 6-inch diameter pipe with an area, A_1, of $0.1963\ ft^2$, the velocity is given as

$$V_1 = Q_1 / A_1 = 6.81 \text{ ft./sec.} \qquad (4.38)$$

If 10% of the flow is drawn off, then $Q_2 = 0.9\ Q_1 = 1.203\ ft^3 / sec.$ If we make the injector pipe 2 inches in diameter and the diameter at the restriction, 4 inches, then the area remaining for the residual main flow around the injector pipe can be found

$$A_2 = A_4 - A_3 = 0.0655\ ft^2 \qquad (4.39)$$

Thus, $$V_2 = Q_2 / A_2 = 18.37\ ft^2/sec. \qquad (4.40)$$

Substituting Eq.(4.38) and (4.39) into (4.37) gives

$$P_1 - P_2 = 287.4\ lb/ft_2 \cong 2 \text{ psi} \qquad (4.41)$$

where $\rho = \gamma/g = 1.563$ lb-sec^2/ft^4
since $\gamma \cong 50$ lb/ft^3 for most fuels.

Thus, the dimensions chosen for the jet pump are suitable since the SEN-DURE heat exchanger requires 2 psi to drive a flow of 60 GPM through it.

Eq. (4.37) can also be written

$$P_1 - P_2 = \rho V_1^2/2\,[\,(V_2^2 / V_1^2) - 1] = \rho V_1^2/2\ (7.982) \qquad (4.42)$$

At $A_3 = 0.0218\ ft^2$, the velocity of the fluid, V_3, with 10% of the overall flow is $6.13\ ft^2/sec.$.
At $A_4 = 0.0873\ ft^2$, the velocity of the fluid, V_4, with 100% of the flow is $15.31\ ft^2/sec.$

In the mixing zone, the momentum equation can be written as

$$(P_2 - P_4)A_4 = \rho Q_4 V_4 - [\,\rho Q_2 V_2 + \rho Q_3 V_3\,] \qquad (4.43)$$

which can be manipulated to give

$$P_2 - P_4 = \rho\ (Q_4/A_4)\ V_4\ \{\ 1\ -\ [\ (Q_2/Q_4)(V_2/V_4)\ +\ (Q_3/Q_4)(V_3/V_4)]\}$$

$$= \rho\ V_4^2\ [\ -0.12\ ]\ = \rho\ V_1^2/2\ (2)(V_4/V_1)^2\ (-0.12)$$

$$= \rho\ V_1^2/2\ (\ -1.21) \qquad (4.44)$$

when the ratios of the various flow rates and velocities above are used.

The Bernoulli Equation can be written between points 4 and 5 as follows,

$$P_4 - P_5 = \eta\,(\rho/2)\,(V_5^2 - V_4^2) \tag{4.45}$$

where η = diffuser efficiency

Realizing that $V_5 = V_1$, Eq.(4.45) can be rewritten as

$$P_4 - P_5 = \rho V_1^2/2\,[\eta(1 - V_4^2/V_1^2)]$$

$$= \rho V_1^2/2\,(-2.64) \tag{4.46}$$

if we use the correct velocity ratio and take the diffuser efficiency, η, at a typical value of 0.65 for this geometry.

Combining Eqs.(4.42), (4.44), and (4.46) gives

$$P_1 - P_5 = (P_1 - P_2) + (P_2 - P_4) + (P_4 - P_5) = \rho V_1^2/2\,(4.132)$$

$$= 148.8\ \text{lb/ft}^2 = 1.03\ \text{psi} \tag{4.47}$$

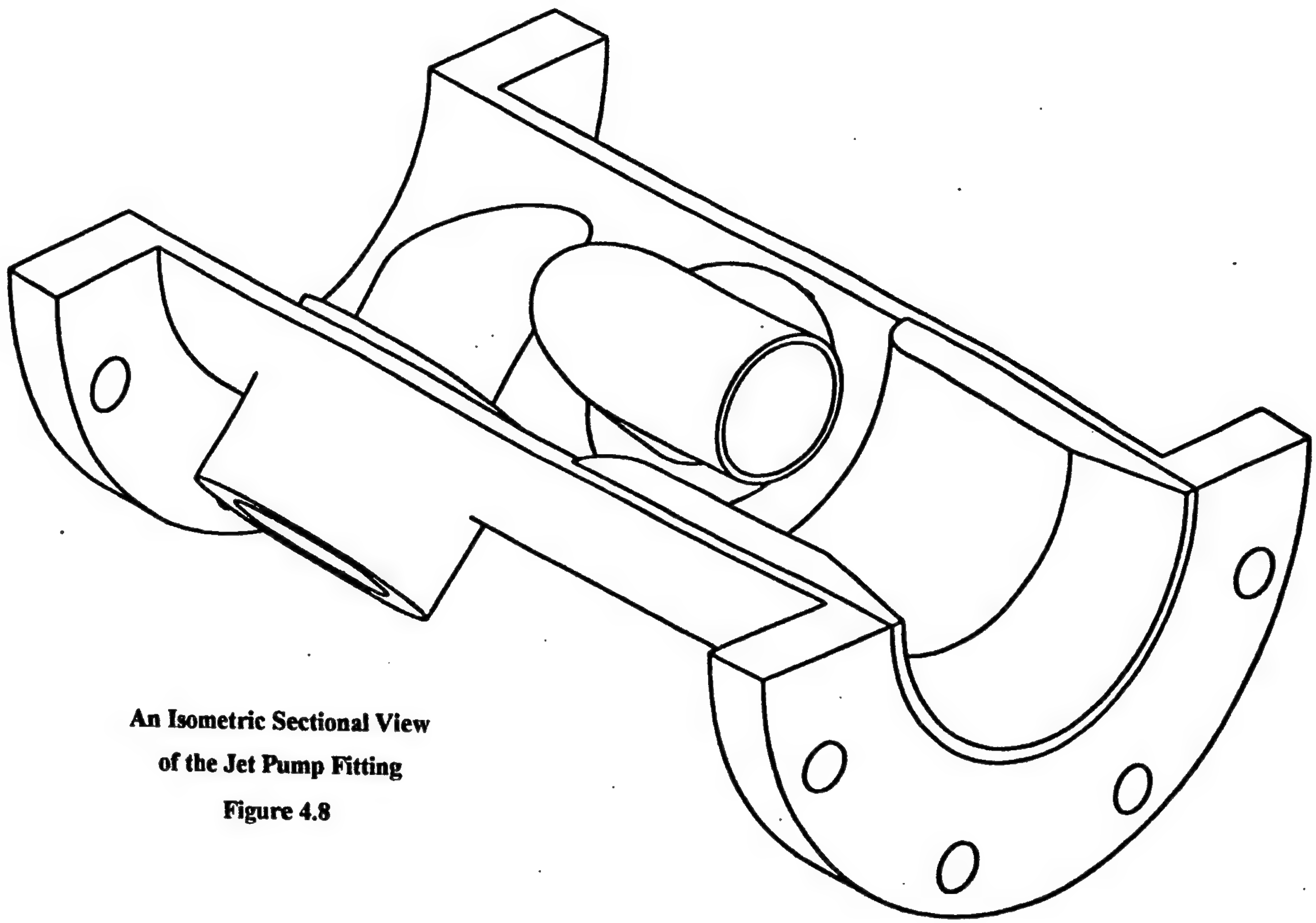
This pressure loss should not be a problem for the system.

A special fitting to implement this concept in the AMBP system has been designed and is described in Dwg. 96008-024. A sectional rendering of the fitting is shown in Figure 4.8.

At flowrates other than the design flowrate, the cooling flow will drop accordingly. In fact, since the pressure drops through both the pumped flow line and the heat exchanger are proportional to the square of velocity, the flow through the heat exchanger will drop off in proportion to the main flow. The power required of the engine is proportional to the product of the velocity and the pressure drop in the line. Thus, the power will drop according to the cube of velocity or flowrate. However, we do not have any data as to the engine cooling requirements at these low power levels.

For example, at 200 GPM, assuming that the pipeline characteristics remain the same, the pressure rise required from the pump will be 1/9th that required at 600 GPM. Thus, if the pump were operating at the maximum design point of 153 psi. at 600 GPM, only 17 psi. would be required at 200 GPM. If we examine the pump characteristic curves at 200 GPM and 17 psi., we can conclude that the engine will be operating at idle speed under these conditions and, if the engine produced a total power of 89 HP at 600 GPM, it will be required to produce only about 5 HP at the lower flowrate, assuming a steady power requirement of 2 HP independent of flow.

At these low power levels, the engine will certainly require much less cooling capacity. However, since the cooling capacity of most engine cooling systems does not drop off with speed to the extent likely for this system, it is not a usual concern and no data is available at these levels. Thus, this is a subject that will require some testing of the prototype over the range of potential operating conditions.



An Isometric Sectional View
of the Jet Pump Fitting
Figure 4.8

The system can easily be modified, if necessary, to increase the cooling flow by using a sleeve in the jet pump fitting which provides a smaller throat diameter. This will increase the pressure drop at a given total flowrate and, thus, will increase the portion of the flow passing through the heat exchanger. It will also increase the pressure drop in the main line through the jet pump fitting, but this could be increased considerably before it creates a problem for the overall system design.

5. ELECTRICAL DESIGN CONSIDERATIONS

During Phase I, the basic design decisions regarding the method of control and telemetry were made. These decisions required an electrical design, integrated with the mechanical design to impement the control methods.

As a general philosophy, an attempt was made to keep as much of the system as possible on the equipment pallet so that the system can be lifted out intact for maintenance. Thus, the NEMA electrical enclosure which contains most of the electrical components is mounted on a frame which is attached to the pallet. Likewise, the four main batteries are mounted on the pallet. The only electrical components which cannot be mounted on the pallet are the fuel level transmitter, which is on the fuel tank; the bilge pump, which is mounted between the hull I-beams in the bilge; and the beacon strobe light, which is mounted on the outside mast. Also, an RG-58/U antenna cable will run from the electrical enclosure to an antenna on the mast.

Some of the important points regarding specific electrical subsystems are discussed as follows:

(a) Main Power

The main electrical power circuit is shown in Drawing No. 96008-005. The four main batteries are wired in a series parallel arrangement that allows separate 12 and 24 volt feeds to the electrical enclosure. The main power breakers are to be located in a separate enclosure which is accessible through the access panel on the hatch cover. They can, thus, be used as master switches for turning the entire system on an off manually. These breakers and the wires from the batteries to the enclosure are sized by taking into account the electrical loads listed in Tables 5.1 and 5.2.

TABLE 5.1
12 VOLT ELECTRICAL LOAD

COMPONENT	CURRENT DRAIN, amps
Bilge Pump	5
Radio Modem (operating at 15 watts)	6
Total	11

TABLE 5.2
24 VOLT ELECTRICAL LOAD

COMPONENT	CURRENT DRAIN, amps
Fuel Level Transmitter	0.5
Barber Colman System	25
Ball Valve Actuator	10
Butterfly Actuator #1	16
Butterfly Actuator #2	16
Magnetrol Fuel Valve	1.2
Opto 22 System	2
Electrical Enclosure Heater	5
Strobe Beacon	0.5
3PDT Relay	0.05
Ignition Relay	0.05
Total	76.3

The rating selected for the 12 VDC circuit breaker is the closest available to 11 amps or 13 amps. The wire size selected is 14 AWG with insulation rated for 90°C. which provides a current rating of 19.25 amps at 135°F.(12)

The rating selected for the 24 VDC circuit breaker is 80 amps which is closest available rating for the model selected to the total load of 76 amps. The wire size selected is 4 AWG with insulation rated for 90°C. which provides a nominal rating of 97 amps and a rating of 74.7 amps at 135°F.(12)

The total electrical load listed in Tables 5.1 and 5.2 is about 1960 watts at 24 VDC or corresponds to about 80 amps at that voltage. The alternator supplied with the engine is rated for 40 amps or 960 watts. Thus, it can be seen that, in the unlikely event that all electrical demands were made at once including all three large valves and the engine speed control actuator, there would be a deficit required from the battery of about 40 amps beyond the current supplied by the alternator. Since the specified batteries will have a total cold-cranking rating of about 2400 amps at 0°F., there should be no problem supplying this deficit for reasonable periods of time. For example, the rated reserve capacity of these batteries will provide 50 amps for a period of 235 to 465 minutes at 80°F.(13)

In the electrical enclosure, itself, the 12 VDC and 24 VDC main cables are terminated in busses for these two voltages. Common or ground for these two voltages is the metal pallet itself, and is a negative ground in terms of battery polarity.

Each of the major components in the enclosure is then powered from the appropriate buss through a circuit breaker with an appropriate rating. This also provides the means to shut down individual circuits during maintenance or trouble shooting. Entrelec, Series GMU-X-U Din Rail Mounted circuit breakers were specified. However, any similar breakers which are rated for the appropriate DC currents and can be conveniently mounted in the enclosure can be used.

(b) Bilge Pump

The wiring of the bilge pump is shown on Drawing 96008-005. We have specified a Whale Model SS5112 pump but any bilge pump with a self-contained level switch that fits between the structural I-beams of the hull and does not extend above the top of these beams can be used.

A connector is provided so that the pump can be disconnected prior to lifting out the machinery pallet for service. This connector should be installed near the electrical enclosure for access from above.

The 12 VDC output from the level switch is tapped and run back to an OPTO 22 Input module via the connector so that the ON-OFF condition of the bilge pump can be monitored from the shore-based station.

(c) Engine Speed Control

Barber Colman speed control hardware was selected for the engine. This system is basically an analog feedback control system that senses engine speed and controls the engine speed governor to hold an set-point value which is provided to the system as an input. For example, in a conventional system, this input signal would be provided by a speed control potentiometer which acts as an electrical throttle.

For the AMBP system, wired as shown in Drawing 96008-007, the input "set-point" signal is provided by the on-board OPTO 22 computer. This signal is derived by the computer from an algorithm which responds to measured pump suction and discharge pressures by reducing engine speed from the nominal value if the pressures deviate from the set values.

The wiring arrangement and components specified in Drawing 96008-007 uses a 4-20 mA signal as the input set-point signal from the computer. It accordingly uses a Barber Colman current controller and a conventional engine speed controller and actuator. The wiring arrangement follows the suggested circuit provided by Barber Colman for this purpose.

The speed input to the engine speed controller is derived from a magnetic speed pickup signal used by the engine tachometer. It is possible that the output from this transducer will be found to be inadequate to drive both the tachometer and the engine speed controller. In that case, a separate magnetic speed pickup will be required on the engine for the speed control system.

The wiring arrangement of Drawing 96008-007 provides for a MANUAL-AUTOMATIC switch which should be mounted on the outside panel of the electrical enclosure along with its associated 10-turn potentiometer. When the switch is in the manual mode, the engine speed can be controlled manually by means of the 10-turn pot. When it is in the automatic mode, the computer will control the speed to be at the set speed provided from the shore station minus any speed decrement that the computer requires to lower the discharge pressure or raise the suction pressure to within set limits.

(d) Engine Instrumentation

The engine is supplied with a remote instrument panel containing gauges, ignition keyswitch, and start button. This panel should be mounted on the AMBP electrical enclosure panel. It can then be used to start and monitor the engine when it is run manually from outside the hull.

As shown in Drawing 96008-008, the panel is coupled to the on-board computer system for remote operation. the key-switch and start button are wired in parallel to relays which are controlled, in turn, by OPTO 22 output modules. Thus, the ignition can be turned on and the engine started from the remote shore station.

The tachometer input leads are tapped at the instrument panel and fed to both the Barber Colman Engine Speed Controller (Dwg. 96008-007) and the OPTO 22 RTU which sends the signal to the shore station for display.

The engine temperature and oil pressure transducers used to drive the panel gauges are special purpose transducers which cannot easily be adapted or calibrated to provide a signal for the OPTO 22 telemetry system. Thus, an auxiliary oil pressure gauge mounted with the Cummins transducer on a common manifold will be used. The auxiliary gauge will drive an OPTO 22 input module directly. Likewise, a separate thermocouple will be mounted on the surface of one of the engine coolant nozzles on the engine heat exchanger to monitor the engine coolant temperature. This transducer will be fed directly to an appropriate input module on the OPTO 22 unit.

(e) Compartment Temperatures

The engine compartment will be monitored by a thermocouple mounted in a strategic location on one of the metal surfaces. A location will be selected which is away from any major heat sources and not in direct radiation view of a hot surface.

The electronics, especially the OPTO 22 system, will not operate at the lower end of the specified environmental temperature range. Thus, when the system is operational in cold climates, the electrical enclosure must be heated to maintain a temperature of at least 0°C (32°F) or above. This will be accomplished with a Watlow controller and heater as shown in Drawing 96008-008. The sensing thermocouple in this case should be mounted within the electrical enclosure on a surface near the OPTO 22 units. The temperature controller has a manual knob control and an LED display that shows the set temperature. This can be mounted on the panel of the electrical enclosure so that it can be operated and read from outside the panel.

(f) Valve Actuators.

The valve actuators are wired as shown in Drawing 96008-006. The actuators are supplied by the valve manufacturers for their respective valves for 24 VDC operation. The actuators are powered directly from the 24 VDC buss through appropriate circuit breakers. A 3PDT relay is used in the case of the ball valve to reverse the operation of the valve at each end of the stroke.

The butterfly actuators have separate "open" and "close" controls. In addition, each valve actuator provides switch closures or a voltage signal to indicate the position that it is in. These are wired to the OPTO 22 unit for telemetering to the shore station.

The magnetrol fuel fill valve is simply operated by the OPTO 22 unit which closes a switch on the negative or ground side of the valve coil. Since this valve is only operated manually by an operator at the shore station who can monitor the fuel flow into the tanks, it was not felt that a positive open or closed signal was needed.

6. ROLL STABILITY OF THE HULL

One of the major characteristics of interest for the AMBP hull is its overturning resistance. Because the nature of the overall design requires that the hull be narrower in beam than its overall length, the critical overturning characteristic is its tendency to roll when an overturning moment is encountered.

As the design of the hull progressed during this phase of the work, the shape of the hull gradually evolved from a simple rectangular box to the final boat-like hull. Roll stability analyses were performed as this design evolved and, in fact, resulted in some of the later design modifications of overall beam, superstructure height, and hatch material. Also, during this process, it was found necessary to relocate the batteries in the compartment to a low position and reposition some of the heavier valve elements inside the compartment.

In this section, the final roll stability characteristics are determined for the final hull configuration shown schematically in Figure 6.1.

It is first necessary to determine the location of the waterline on the hull. This is accomplished by calculating the displacement of the various parts of the hull. The volume displaced by the triangular pyramid abb'd is given by

$$V_1 = 1/3\ A_1 h = 756 \text{ in.}^3 = 0.44 \text{ ft.}^3 \qquad (6.1)$$

where $h = 18$ inches

and
$$A_1 = 1/2(ab' \times ad \cos \alpha) = 126 \text{ in.}^2 \qquad (6.2)$$

$$\alpha = \tan^{-1} 45/24 = 61.9°$$

The volume displaced by the rectangular pyramid bb'dcc' is given by

$$V_2 = 1/3\ A_2 h' = 4968 \text{ in.}^3 = 2.88 \text{ ft.}^3 \qquad (6.3)$$

where $h' = 24$ inches

and
$$A_2 = 18 \times 34.5 = 621 \text{ in.}^2 \qquad (6.4)$$

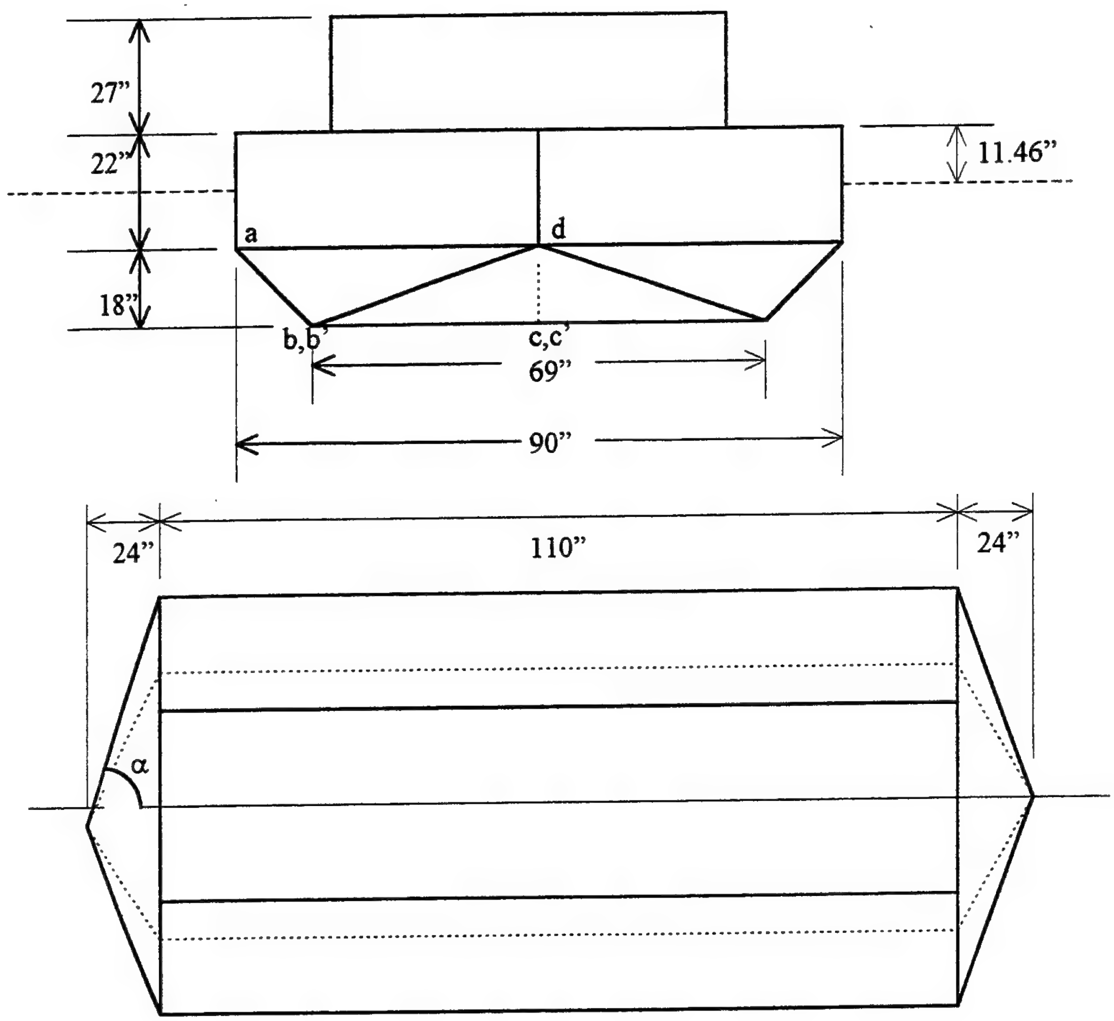
The volume displaced bythe central portion of the lower hull is given by

$$V_3 = 110 \times A_3 = 157{,}410 \text{ in.}^3 = 91.1 \text{ ft.}^3 \qquad (6.5)$$

where
$$A_3 = 18 \times 1/2\ [\,90 + 69] = 1431 \text{ in.}^2$$

The total volume displaced by the lower hull is given by

$$V_L = 4(\,V_1 + V_2) + V_3 = 104.38 \text{ ft.}^3 \qquad (6.6)$$



Schematic Diagram of the Hull
Showing Dimensions and Notation
Figure 6.1

The corresponding weight of water displaced by the lower hull is

$$W_L = 104.38 \times 64 = 6680 \text{ lbs.}$$

The total water displaced by the upper hull section (not including the superstructure or coaming above the deck) is given by

$$V_U = A_D \times 22/12 = 153.54 \text{ ft.}^3 \qquad (6.7)$$

where the deck area is given by $A_D = [\,90 \times 24\,] + [\,90 \times 110\,] = 12{,}060 \text{ in}^2$

$$= 83.75 \text{ ft.}^2 \qquad (6.8)$$

Thus, the weight of the water displaced by the upper hull

$$W_U = 153.54 \times 64 = 9827 \text{ lbs.} \qquad (6.9)$$

Thus, the total displacement of the hull is

$$W_D = W_L + W_U = 6680 + 9827 = 16{,}507 \text{ lbs.} \qquad (6.10)$$

The current estimated total weight of the hull and its contents, W, has been determined from the CAD design of the entire system and is 11,390 lbs. Thus the reserve buoyancy of the AMBP is the difference between these weights or

$$B = 5117 \text{ lbs.} \qquad (6.11)$$

The freeboard of the hull (distance from the deck to the waterline) can be found from

$$f = \frac{B}{64 \times A_D} = 0.954 \text{ ft.} = 11.46 \text{ inches} \qquad (6.12)$$

Thus, the waterline is predicted to be 11.46 inches from the deck or 28.54 inches from the "keel".

The location of the center of buoyancy from the keel can be determined from the relationship

$$b = \frac{W_L b_1 + W'_U b_2}{W} = 14.9 \text{ inches} \qquad (6.13)$$

where b_1 is the distance from the keel to the c.b. of the lower hull which is approximately 9 inches, b_2 is the distance from the keel to the c.b. of the portion of the upper hull below the waterline which is found from

$$b_2 = 18 + (22 - 11.46)/2 = 23.27 \text{ inches} \qquad (6.14)$$

and W'_U is simply the displacement of that portion of the upper hull below the waterline, 4710 lbs, which can be found by subtracting W_L from the total weight, W.

The metacenter which is the apparent center of rotation for the hull in roll can be located from the relationship (2)

$$b_m = \frac{I}{V} \qquad (6.15)$$

where the moment of inertia of the hull, I, is given by

$$I = 1/12 \;\; A_D\, w^2 = 392.6 \text{ ft}^4, \qquad (6.16)$$

A_D is given in Eq. (6.8) and the width of the hull, w, is 7.5 ft.

The volume of the water displaced, V, is the weight of the water, 11,390 lbs. divided by 64 or 177.96 $ft.^3$ Substituting these values into Eq.(6.15) gives

$$b_m = \; 392.6/177.96 = 2.206 \text{ ft.} = 26.47 \text{ inches} \qquad (6.17)$$

Thus, the metacenter is located at 14.9 + 26.47 = 41.37 inches from the keel.

The c.g. of the AMBP was estimated using data from the detailed CAD design of the system as lying at 27.8 inches from the keel and along the centerline. The metacenter is, thus, 13.57 inches or 1.13 ft. above the c.g. It is necessary, of course, that the metacenter be located above the c.g. if the system is to be stable in roll.

The degree of stability can be estimated by realizing that the righting moment,M, for small angles of roll, θ, measured in radians can be found from

$$M = W\, d_{gm}\, \theta = 12{,}871\; \theta \;\; \text{ft-lbs.} \qquad (6.18)$$

Table 6.1 gives values for the righting moment for roll displacements from one to 15 degrees. A 200 lb. man standing at the very edge of the deck would create a capsizing moment of about 750 ft.-lbs. Thus, the hull would reach an equilibrium angle of about 3.33 degrees in this case. Another way of expressing the stability is that it would take more than four 200 lb. men standing on the same side to roll the hull 15 degrees.

TABLE 6.1
RIGHTING MOMENTS CORRESPONDING TO VARIOUS ROLL ANGLES

θ, degrees	θ, radians	M, ft.-lbs.
1	0.01745	224.64
5	0.08725	1123.2
10	0.1745	2246.4
15	0.26175	3369.6

7. MOORING SYSTEM DESIGN

7.1 SELECTION OF CONFIGURATION

Early in the program, three promising mooring concepts were identified and analyzed in a preliminary fashion using a simple box-like hull design concept which appeared promising at that time. The analysis was included as an appendix in Status Report #4 for this program. (3) Because this preliminary hull design was abandoned, the actual numerical results of the analysis are not significant. However, the relative comparison of line tensions and the qualitative comparisons of the relative advantages and disadvantages among the three systems is meaningful and allows the selection of the most promising system to be carried further into the detailed design.

The three mooring concepts are shown schematically in Figure 7.1. For Case I, the hull is moored with a 3/4-inch chain, making a 45° angle with the bottom. All cases are assumed to be in 60 ft. of water. In addition to the chain, two NATSYN elastic lines, one on each end of the hull are added to prevent rotation.

For Case II, the hull is moored with two 3/4-inch chains, one on either end of the hull. At the design current of 2.5 knots, the upstream chain is made to be tangential to the bottom.

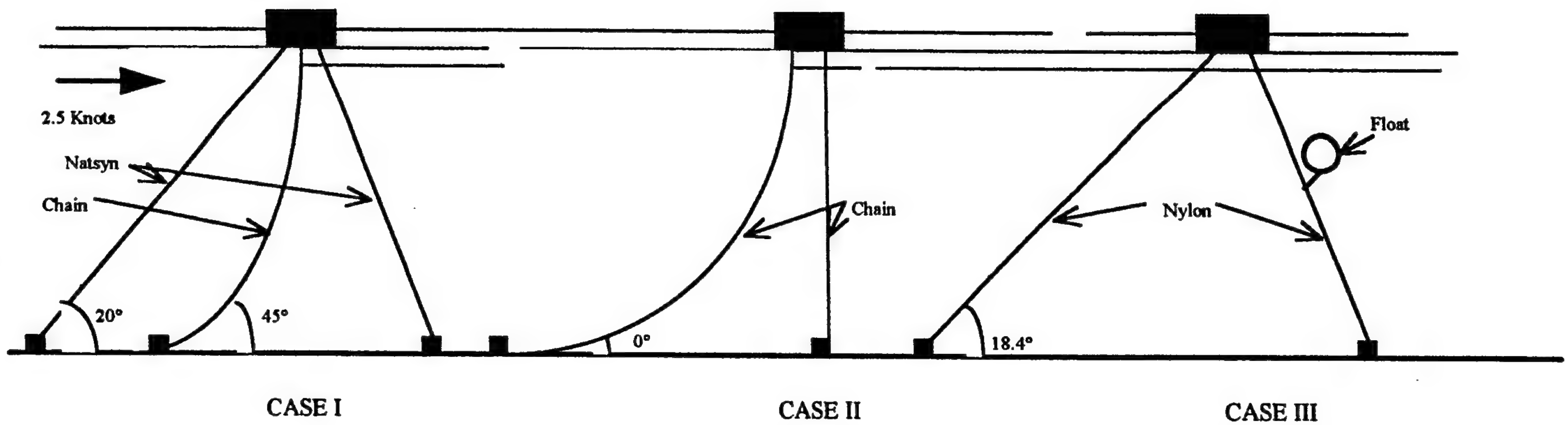
For Case III, the hull is moored with one inch NYLON lines on each end. At the design current, the upstream line makes an 18.4° angle with the bottom.

The results of the numerical calculations of mooring line forces and hull movements are shown in Table 7.1. Again, these should be viewed as relative comparisons. No importance should be placed on the absolute values of the variables.

TABLE 7.1
RELATIVE PERFORMANCE OF THE THREE MOORING CONFIGURATIONS

CASE #	MOORING LINE MAT'L	MOORING LINE TENSION, LBS.						HULL CHARACTERISTICS		
		AT THE HULL			AT THE ANCHOR			DRAFT	FREEBOARD	PITCH
		HOR.	VERT.	TOTAL	HOR.	VERT.	TOTAL	FT.	FT.	DEG.*
I	3/4" chain and NATSYN	1450	1868	2365	1450	1450	2050	4.49	0.95	22.4
II	3/4" chain	1200	900	1500	1200	-	1200	4.13	1.31	15.0
III	1 " NYLON	1135	375	1195	1135	375	1135	3.7	1.7	13.5

*The pitch angle for these cases is such that the stern or the downstream end is lower in the water.



Schematic Diagram of Three Mooring Concepts

Figure 7.1

Some of the advantages and disadvantages of each of these cases are discussed as follows:

Case I

The principal advantage of Case I is to require only one main mooring line and one main anchor. Moreover, since the main mooring line is attached at the center of the hull, the tension should have a small effect on the pitch angle. However, the single anchor requires stabilizing NATSYN lines to prevent spin or requiring the hoseline to prevent spin. To limit the elongation of the NATSYN line, the chain must be relatively short which results in a large tension. This, in turn creates a large draft, a small freeboard, and a large drag force.

Moreover, Case I introduces the need for two additional anchors to secure the NATSYN line and greater deployment difficulty to deploy three legs.

Case II

The main advantage of Case II is to provide zero vertical component of tension at the anchor, enabling the use of embedment anchors. Thus, the components will be familiar to operating personnel and deployment will be straightforward. The chain, however, is heavy, and the tension at the hull can be fairly large. The higher tensions with the chain attached at the "bow" can produce added pitch which may be a problem under survival current conditions. Also, the chain will pile up on the downstream side.

Case III

The principal advantage of NYLON is its light weight in water. The weight will not add to the tension. The downstream leg will not pile up on the bottom. (The small weight of the line can be supported by a small float in the middle of the line.)

If the line is long enough, the angle at the bottom can be kept small, resulting in a small vertical force on the anchor. Thus, like Case II, the lines and anchors will be familiar and deployment will be straightforward. Both Cases II and III, of course, require at least two main anchors.

Conclusions

Of the three cases, Case III, the NYLON mooring line configuration, appears to offer the lowest total mooring line forces, both at the anchor and at the hull attachment. It also yielded the smallest draft and pitch angle. Although it may be only marginally better than Case II, the NYLON lines are expected to be lighter and easier to handle than the chain and the response of the mooring, including stretch, will be easy to predict. The Case III mooring will be familiar to Navy personnel and will be straightforward to deploy.

This preliminary analysis did show a surprising result concerning the hull pitch angle which was stern heavy or down for all three cases. This shows the importance of carefully evaluating the significance on the hull of all of the external forces and selection of the location at which to apply the mooring force. It should be attempted to reduce the net pitch angle of the hull under the applied design and survival currents to below 5 degrees.

On the basis of this preliminary analysis, it was decided to proceed with a mooring design using two identical NYLON mooring lines, one in the bow and one in the stern. The detailed configuration will be developed in the following subsections of this report.

7.2 MOORING LINE TENSION

The final shape and principal dimensions of the hull were given in Figure 6.1. The objective of the analysis of this subsection is to obtain reasonable estimates of the tension in the upstream mooring line under a design current of 2.5 knots and a survival current of 5 knots.

The tension force in a mooring line can be written as

$$T_M = \sqrt{R_H^2 + R_V^2} \qquad (7.1)$$

where R_H is the resultant of the horizontal forces applied to the hull moored in a given current and R_V is the resultant of vertical forces acting on the hull.

The free body diagram of the hull is shown in Figure 7.2. The forces acting on the hull are:

- D_B, the drag force induced by the current acting on the hull bow.
- T_M, the tension in the mooring line.
- B, the weight of the water displaced.
- W, the weight of the hull, including payload.
- T_P, the horizontal component of the tension in the hoseline = 500 lbs for 2.5 knots and 2000 lbs. for 5 knots (see Table 3 of Page 36 of Ref. (1)

The important angles are:

- θ, the angle between the mooring line and the horizontal, in the vertical plane.
- α, the angle between the hoseline and the longitudinal axis of the hull, in the horizontal plane. The analysis of Page 36 of Ref.(1) showed that this angle would be 37.8 degrees.

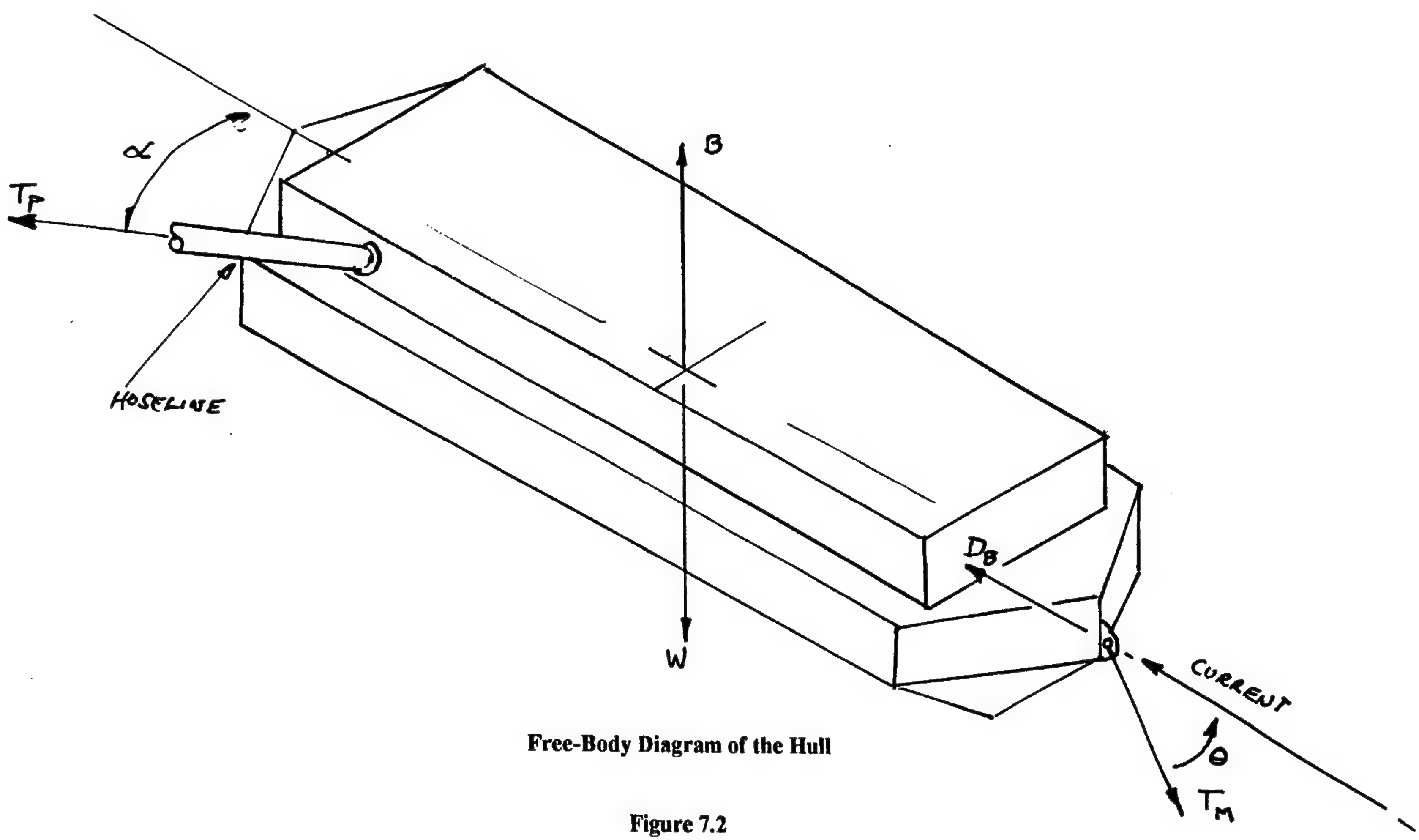
(a) Estimating the Draft

The principal dimensions of interest were shown in the schematic diagram of Figure 6.1. The volume of sea water displaced by the hull is the sum of the displacement of the lower hull and the displacement of the submerged part of the upper hull.

From Eq. (6.6), the volume of the lower hull is 104.38 ft.^{3}. The submerged volume, V, of the upper hull is given by

$$V = (d - 1.5)\ 83.75 = 83.75\ d - 125.625 \qquad (7.2)$$

where the deck area., previously calculated in Eq. (6.8), of 83.75 ft.^{2} is used.



Free-Body Diagram of the Hull

Figure 7.2

Case I - Design Current (2.5 knots = 4.1ft./sec.)

Let the stretched length of the mooring line (assuming a 10% elongation) be

$$L = 1.1[(3D)^2 + D^2]^{1/2} = 3.4785\ D \qquad (7.3)$$

Then $\sin\theta = D/3.4785D$ or $\theta = 16.7$ degrees

An estimated value of the mooring line tension could be 1200 lbs. for the design current of 2.5 knots (4.1 ft./sec.) The vertical component of tension would then be

$$T_{MV} = T_M \sin 16.7 = 345 \text{ lbs.} \qquad (7.4)$$

The resultant horizontal external force applied to hull is

$$R_H = D_B + 2T_P \cos\alpha \qquad (7.5)$$

The resultant vertical force is simply

$$R_V = B - W \qquad (7.6)$$

Adding the upper hull displacement of Eq.(7.2) to the lower hull displacement of 104.38 ft.^{3}, multiplying by the density of water and equating to the sum of the buoy weight and the vertical component of mooring line tension given by Eq. (7.4) gives

$$(104.36 + 83.75d - 125.625)\ 64 = 11{,}390 + 345$$

or

$$d = 2.44 \text{ ft.} = 29.3 \text{ inches} \qquad (7.7)$$

Thus, the new waterline is 29.3 inches from the keel or 10.7 inches from the deck. It can be seen that the hull has moved down in the water approximately 3/4 inch as a result of the mooring line tension.

The drag area of the bow, or stern, of the hull can now be calculated, knowing the wetted, projected area. This is a straightforward exercise in geometry which will not be repeated here. However, the resulting cross-sectional or projected area is 17 ft.^{2}

The drag force imparted by a current can be found from the relationship

$$D_B = \tfrac{1}{2}\rho\ C_D A V^2 \qquad (7.8)$$

where
ρ = sea water density = 2 slugs/cu.ft.
V = current velocity, ft./sec.= 4.1 ft./sec. for 2.5 knots
A = projected area of the hull normal to the current = 17 ft.^{2}
C_D= drag coefficient of the hull.

Experience teaches that a typical drag coefficient for a blunt hull of this type would be expected to lie between 1.0 and 1.2. Using the latter value and substituting into Eq. (7.8) gives

$$D_B = 343 \text{ lbs.} \tag{7.9}$$

The horizontal component of mooring line force can be written

$$T_{MH} = 2T_p\cos\alpha + D_B = 1133 \text{ lbs.} \tag{7.10}$$

for the numerical values given above. The tension in the mooring line is then found from the relationship

$$T_M = T_{MH}/\cos\theta = 1183 \text{ lbs.} \tag{7.11}$$

which is fairly close to the assumed value of 1200 lbs. If this value differed substantially from the assumed value, it would be necessary to iterate the solution by assuming a new value of mooring line tension, recalculating the vertical force, draft, and drag forces. The horizontal component would then be used to recompute the resultant tension and compared with the assumed value. The iteration would be repeated until the calculated value converges with the assumed value. However, in our case, the calculated value is close enough for practical purposes without iteration.

Case II - Survival Current (5 knots = 8.2 ft./sec.)

For survival current, we will let the elongation be 15% resulting in a stretched length (using an equation analogous to Eq. (7.3)) of 3.637D which, in turn results in the mooring line angle

$$\theta = 16 \text{ degrees} \tag{7.12}$$

The assumed tension for this current will be 5000 lbs. The vertical component of tension, using Eq. (7.4) with an angle of 16 degrees, becomes 1378 lbs. Similarly, the vertical resultant force becomes

The revised draft becomes $d = (12768 + 1360)/ 5360 = 2.64 \text{ FT.} = 31.6 \text{ inches}$ (7.13)

Thus, the new waterline is 31.6 inches from the keel or 8.4 inches from the deck.

The revised drag area for this new draft becomes 18.44 ft.^{2}

If we still assume a drag coefficient of 1.2, Eq. (7.8) with the above drag area and a velocity of 8.2 ft./sec. gives an increased drag force

$$D_B = 1488 \text{ lbs.} \tag{7.14}$$

The horizontal component of mooring line force can then be found from Eq.(7.10), remembering that the hoseline force is 2000 lbs. for this current velocity. The result is

$$T_{MH} = 4648 \text{ lbs.} \tag{7.15}$$

The total mooring line tension is then found from Eq. (7.11) with the new value of θ,

$$T_M = 4835 \text{ lbs.} \tag{7.16}$$

which is close enough to the assumed value of 5000 lbs.

The results of the mooring line tension analysis can be summarized as shown in Table 7.2, where the component forces are rounded up values.

TABLE 7.2
MOORING LINE TENSION FOR DESIGN AND SURVIVAL LEVELS OF CURRENT

CURRENT KNOTS	HORIZONTAL TENSION, LBS.	VERTICAL TENSION, LBS.	TOTAL TENSION, LBS.
2.5	1150	350	1200
5.0	4800	1400	5000

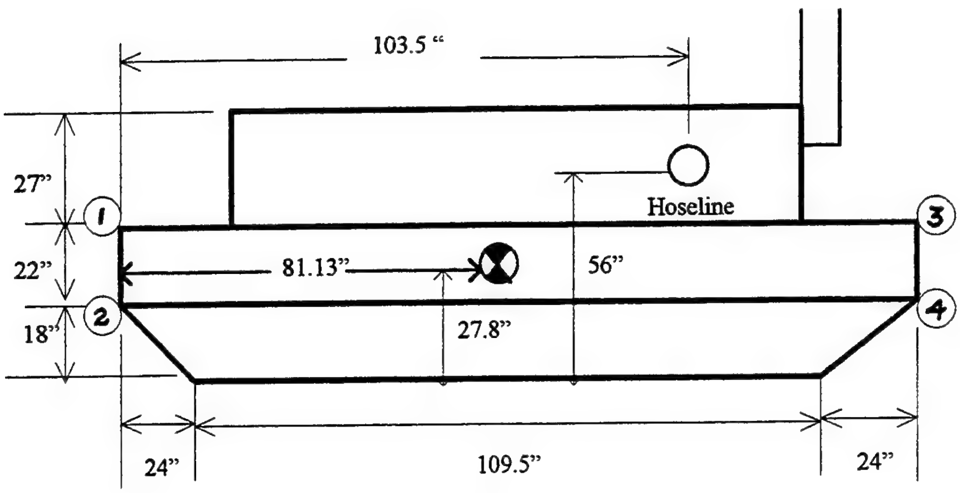
7.3 SELECTION OF THE POINT OF ATTACHMENT

Knowing the force levels produced in the mooring line by the current, the next task is to select the point of attachment on the hull for the mooring lines. Two possible locations on each end of the hull were selected for convenience. The basis upon which the selection can be made is the pitch characteristics produced by the mooring location. This is a similar moment stability question to that addressed for roll in Section 6. except that here, the angular motions are about a tranverse axis instead of the longitudinal one.

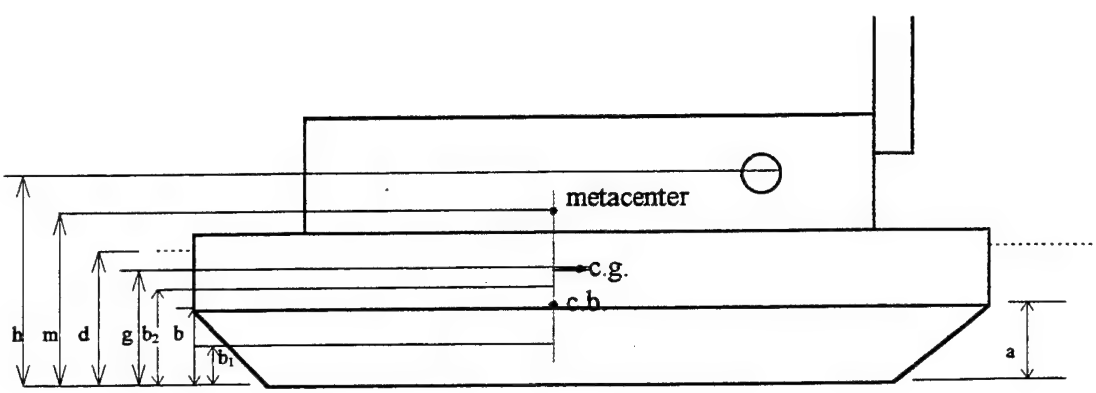
The criterion for selection is that the attachment location should produce minimum pitch angle under design and survival current conditions. Moreover, it is desirable that a bow up attitude should result, if any pitch angle is produced, particularly under the survival current.

For the AMBP design, the hull is not symmetric for pitch calculations since the hoseline connections are off-center. Thus, we have defined the stern as the end of the hull closest to the hoseline and the mast. Four attachment points have been considered as shown in Figure 7.3(a).

1. at the bow and at deck level
2. at the bow and at the junction between upper and lower hull
3. at the stern and at deck level
4. at the stern and at the junction between upper and lower hull



(a) Attachment Points and Principal Dimensions



(b) Nomenclature for Distances from Keel

Schematic Diagram Showing Dimensions and Nomenclature
Figure 7.3

A number of distances from the keel are defined in Figure 7.3(b) as follows:

a = point of attachment of the mooring line.
b = c.b. location for the whole hull
b_1 = c.b. of the lower hull = 9 inches
b_2 = c.b. of the submerged part of the upper hull
d = draft
g = c.g. of the whole hull = 27.8 inches
h = attachment point of the hoseline = 56 inches
m = metacenter

The schematic diagram of Figure 7.4 shows the following forces applied to the moored hull and their points of application. The forces, some of which we saw in Sections 6. and 7.2 are as follows:

T_M, The upstream mooring line tension
T_{MH}, The horizontal component of mooring line tension
T_{MV}, The vertical component of mooring line tension
W, The air weight of the hull and payload = 11,390 lbs.
B, The weight of the water displaced = W + T_V, applied at the c.b.
T_P, The horizontal component of hoseline tension
D_B, drag force due to the current, assumed to apply at the c.b.

When the hull is subjected to these forces, the hull will assume an equilibrium pitch angle, β. The condition of moment equilibrium requires that the righting moment, M_1, about the center of gravity, c.g., resulting from the water presure on the bottom of the hull with a pitch angle, β, and given by

$$M_1 = B(m - g)\sin\beta \cong B(m - g)\beta \qquad (7.17)$$

must be equal and opposite to the resultant moment of the external forces about the c.g. These moments are defined as follows:

M_2, Moment induced by the two hoselines and may be written

$$M_2 = 2t_p\cos\alpha\,(h - g) \qquad (7.18)$$

M_3, Moment induced by the horizontal component of mooring line tension given as

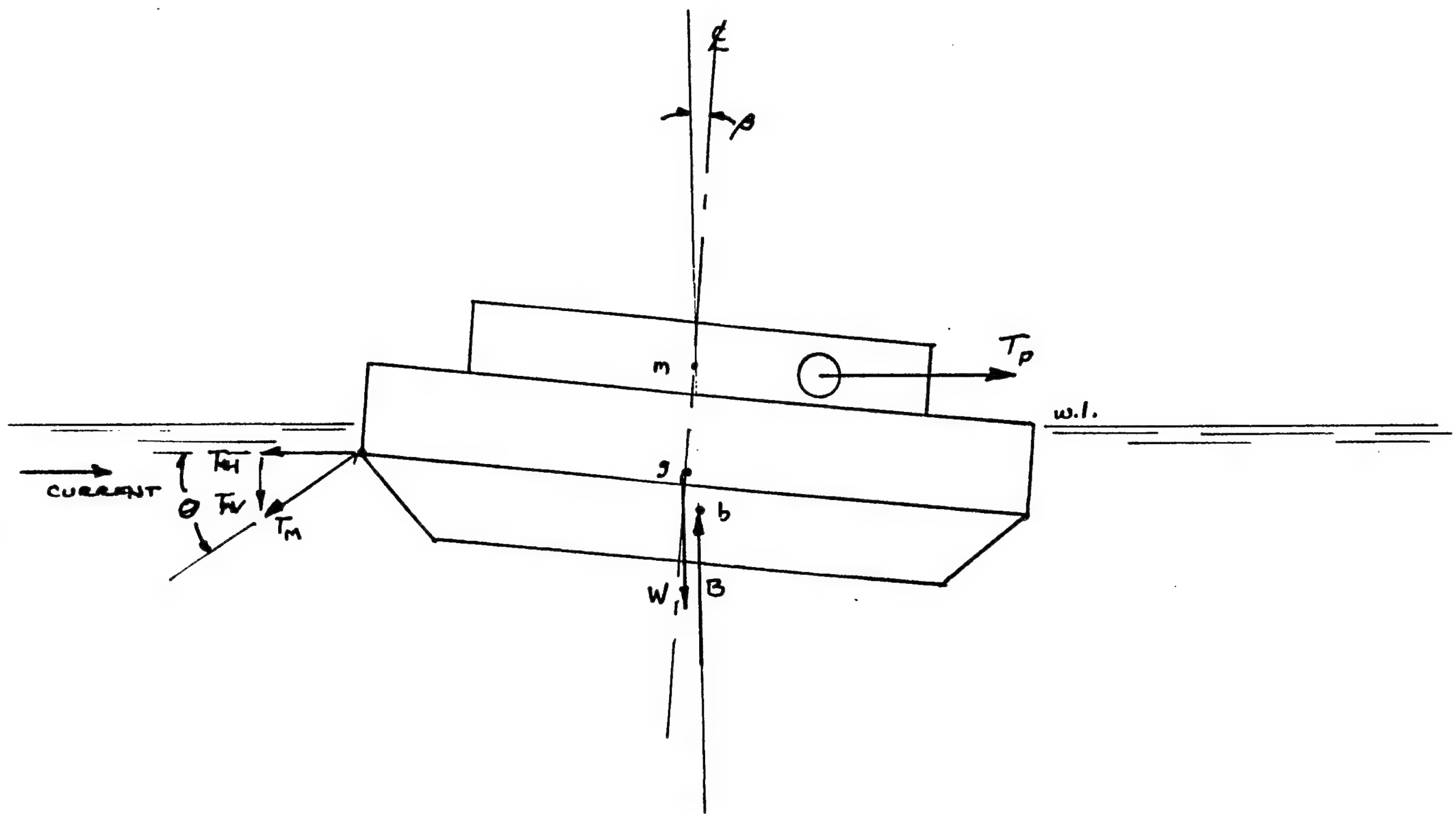
$$M_3 = T_{MH}\,(a - g) \qquad (7.19)$$

M_4, Moment induced by the vertical component of the mooring line tension given as

$$M_4 = T_{MV}\,(81.13) \qquad (7.20)$$

M_5, Moment induced by the drag force on the hull given as

$$M_5 = D_B\,(g - b) \qquad (7.21)$$



Forces Acting on the Moored Hull

Figure 7.4

In order to use the moment equilibrium to calculate the angle, β, we need to determine the location of the c.b. and the metacenter which depends upon the displacement and, in turn, the line tensions.

Realizing that the weight of the water displaced, B, is equal to 64V, Eq. (7.2) can be rewritten as

$$d = (B + 1361)/\,5360 \quad \text{ft.} \tag{7.22}$$

The location of the c.b. above the keel is given by

$$b = \frac{d_1\, b_1 + d_2\, b_2}{d_1 + d_2} \quad \text{inches} \tag{7.23}$$

where d_1 = displacement of the lower hull = 6680 lbs
b_1 = distance from keel to c.b. of lower hull = 9 inches
d_2 = displacement of submerged part of upper hull
$= (12d - 18)(83.75 \times 144)\ 64 \div 1728$ lbs.

simplifying $d_2 = (12d - 18)\ 447$ lbs. (note: d is in ft.) (7.24)

and b_2 = distance from the c.b. of the upper hull to the keel
$= 18 + (12d - 18)/2 = 9 + 6d$ (7.25)

Thus, to find b, the value of d is calculated from Eq. (7.22). This value is used in Eqs. (7.24) and (7.25) to find d_2 and b_2. Finally, these values can be used in Eq. (7.23) to find b.

The distance between the c.b. and the metacenter is given by

$$(m - b) = \frac{I}{V} \tag{7.26}$$

The moment of inertia, I, is given by Eq. (6.16) but, in this case, w is the length of the hull at the waterline instead of the width. Thus, I becomes 1202.3 ft.^4.

The volume of water displaced can be written

$$V = B/64 = (W + T_{mv})/64 \tag{7.27}$$

Substituting these values for I and V into Eq. (7.26) gives

$$(m - b) = 76{,}947/\,(W + T_{MV}) \ \text{ft.} \qquad (7.28)$$

or, we can write $$m = b + 12(m - b) = 923364/(11390 + T_{MV}) \text{ inches} \qquad (7.29)$$

A total of 6 cases were studied including mooring line attachment points at the deck level and at the junction between the upper and lower sections of the hull. Four of the cases were for the design current of 2.5 knots and two for the survival current of 5 knots. The first of these cases was for a current of 2.5 knots flowing into the bow (the end furthest from the hoseline connection) with the mooring line connected at a fitting one inch below the deck level. For this current, the forces acting on the hull were found in Section 7.2 and are summarized as follows:

$T = 1200$ lbs	$B = 11735$ lbs
$T_{MH} = 1133$ lbs	$T_P = 790$ lbs
$T_{MV} = 345$ lbs	$D_B = 343$ lbs.
$W = 11390$ lbs	

Calculating the moments, using Eqs. (7.17) through (7.28), gives the following, taking notice of the direction of the moments and using the convention that positive moments are clockwise in Figure 7.3.

Eq. (7.22) gives	$d = 2.44$ ft.
Eq. (7.24) gives	$d_2 = 5059.7$ lbs.
From earlier analysis	$d_1 = 6680$ lbs.
and	$b_1 = 9$ inches
Eq.(7.25) gives	$b_2 = 23.64$ inches

Substituting these values into Eq. (7.23) gives

$$b = 15.3 \text{ inches}$$

The location of the metacenter above the keel can then be found using Eq. (7.29)

$$m = 94.02 \text{ inches}$$

which corresponds to 66.22 inches above the c.g.

Using the appropriate moment arms and sign convention the above forces give the following moments about the c.g.

$M_1 = 64758\ \beta$ where positive β is taken to be bow down
$M_2 = 1857$ ft-lbs.
$M_3 = -\ 1058$ ft-lbs.
$M_4 = -\ 2333$ ft-lbs.
$M_5 = -\ 357$ ft-lbs.

Setting the sum of these moments equal to zero and solving for β gives

$$\beta = 1891/64758 = 0.0292 \text{ rad.} = 1.67 \text{ degrees}$$

Similar calculations, which will not be shown in detail here, were performed for all six cases and are summarized in Table 7.3

TABLE 7.3
HULL ATTITUDE FOR VARIOUS CURRENT STRENGTH AND MOORING LINE ATTACHMENT POINTS

CASE #	CURRENT FLOW INTO	CURRENT STRENGTH, KNOTS	MOORING POINT OF ATTACHMENT	PITCH ANGLE, DEGS.	HULL ATTITUDE
1	BOW	2.5	DECK	1.67	BOW DOWN
2	BOW	2.5	JUNCTION	0.082	BOW UP
3	STERN	2.5	DECK	1.4	STERN DOWN
4	STERN	2.5	JUNCTION	0.036	STERN UP
5	BOW	5	JUNCTION	0.295	BOW UP
6	STERN	5	JUNCTION	1.41	STERN UP

The conclusions that can be drawn from this analysis is that placing the mooring line attachment point at or near the junction between the upper and lower hulls (i.e., 18 inches from the keel) produces the minimum pitch angle when the hull is subjected to a current. It also produces an "up" attitude for the end into which the current is flowing. This is desirable because the end into the current will be able to resist plowing under the water.

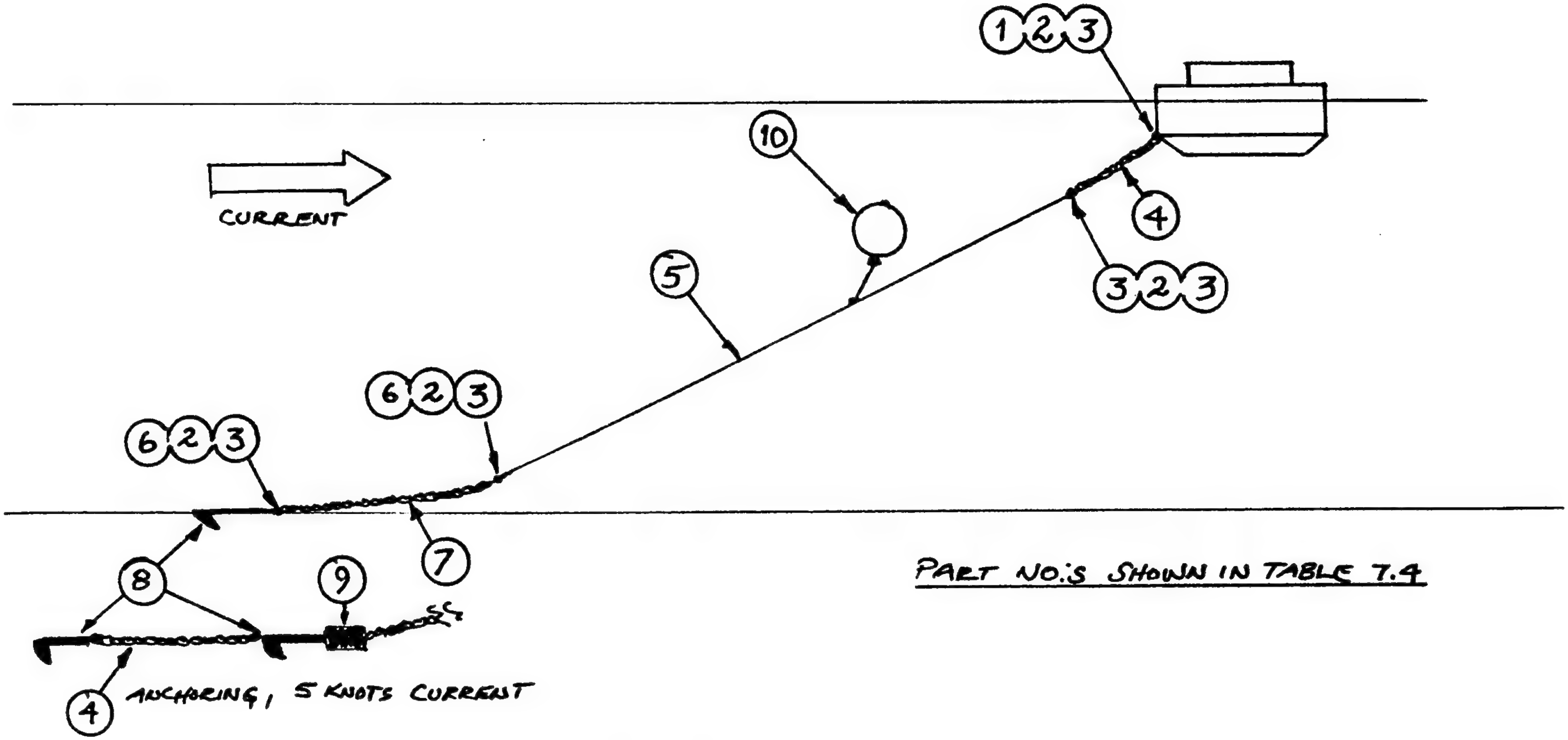
7.4 DETAILED DESIGN OF THE MOORING

The final task for the mooring design is the selection of the various specific components required for the mooring system. This selection was based upon an extensive experience by the designer in a wide range of mooring systems. The rationale for the various selections made is presented in this section as well as a description of the recommended system.

The overall configuration is shown in Figure 7.5. Starting at the hull attachment point, the major components are as follows:

1. A length of chain, long enough to allow for ease in connecting the mooring and to prevent chafing of the Nylon line aganst the hull.

2. A length of Nylon rope, approximately 3 times the water depth.



Overall AMBP Mooring

Configuration

Figure 7.5

3. A length of chain with sufficient length and weight to suppress any vertical component of tension at the anchor point of attachment, under design current conditions. (2.5 knots)

4. An embedment anchor. (2 in 5 knot current)

5. Suitable hardware to connect and disconnect the various components.

The general design condition was taken to be a current of 2.5 knots with provisions specified to increase the anchor holding power to resist the 5 knot survival current.

However, the design tensile load for the mooring components was taken to be 5000 lbs. which is the estimated mooring tension for the survival current of 5 knots.

The major components are specified as follows:

1. Chain

All chains should be Proof Coil, galvanized, welded chain. The section between the hull and the Nylon mooring line and the connection between the two Navmoor anchors should be 5/8-inch chain which has a working load of 6800 lbs., providing a safety factor of 5.5 at 2.5 knots.

The section between the end of the Nylon line and the anchor(s), should be 1-inch nominal. This will provide the weight required to balance the vertical component of line tension, 345 lbs., with a 2.5 knot current with a reasonable chain length.

The upper section of 5/8-inch chain should be about 8 ft. long. The lower section, between the Navmoor anchors for a 5 knot current should be equal in length to the depth of the water.(4)

From the relationships for a catenary, the length of chain, S, required for a chain with a weight per unit length, w, to be tangential with the ground when subjected to a vertical force, T_{MV}, is given by

$$S = T_{MV} / w = 345 / 8.7 = 40 \text{ ft.} \qquad (7.30)$$

for the 2.5 knot current, where the vertical component of tension is 345 lbs. The weight per unit length of 1-inch proof coil chain in water is approximately 8.7 lbs. per foot.

2. Rope

The Nylon rope must be a non-kinking type commonly used for mooring applications. For example, 8 strand plaited type (Cordage Group) or 2 in 1 braided (Samson) would be suitable. Stranded rope should not be used.

A 1-inch nominal diameter rope has a nominal strength of 25,000 lbs. which would provide a safety factor of 5 for the survival current. This is more than adequate and will also provide a comfortable size for handling. Smaller size ropes would be expected to creep under the survival load of 5000 lbs.

The recommended geometry for the mooring system is that the anchor be located approximately 3 times the depth upstream from the hull. Trigonometry suggests, then, that the length of the Nylon

rope be approximately 3.16 times the local depth. For example, if the depth is 60 ft., then the rope should be 190 ft. long.

Both ends of the Nylon rope should be terminated with a 1-inch galvanized thimble and the rope should be spliced back onto itself with a tight and seized splice.

3. Anchors

The best available anchor for this application, upon recommendation from R. Taylor of NCEL is the 100 lb. NAVMOOR anchor which is also one of the specified anchors for the hoseline system which will be used with the AMBP. (5)

The horizontal holding power of this type of anchor can be estimated using the relation

$$H_M = H_R \, (W_A / 10{,}000)^r \quad \text{kips} \qquad (7.31)$$

where H_m is the horizontal holding power

H_R is the baseline capacity of the specified anchor

r is the weight ratio exponent

W_A is the anchor weight in lbs.

Reference (6) gives the following set of values for the NAVMOOR anchor:

H_R = 210 kips for soft clays

= 270 kips for sand

r = 0.94

Substituting into Eq.(7.31) gives

H_M = 2768 lbs. for soft clay

H_M = 3560 lbs. for sand

Since the calculated horizontal force at the anchor for a 2.5 knot design current is 1133 lbs., the NAVMOOR anchor would hold with a 2.4 safety factor in soft clay. However, in the case of a 5 knot current, the horizontal and vertical components of the tension at the anchor are calculated to be 4833 and 1370 lbs., respectively.

Thus, operation in a current of this magnitude would require the following anchoring provisions:

(a) Either a 1000 lb. (in seawater) clump, placed between the 40 feet of 1-inch chain and the 100 lbs. NAVMOOR anchor, or a length of heavier chain to replace the 1-inch chain where the length would be calculated to be

$$S = 1370 / w \qquad (7.32)$$

and w is the weight per foot of this heavier chain.

(b) A second 100 lb. NAVMOOR anchor, connected in tandem, upstream of the first one. A length of 5/8-inch chain equal to the water depth can be used to connect the two anchors. The horizontal holding power will then be approximately 5536 lbs. which is greater than 4833 lbs.

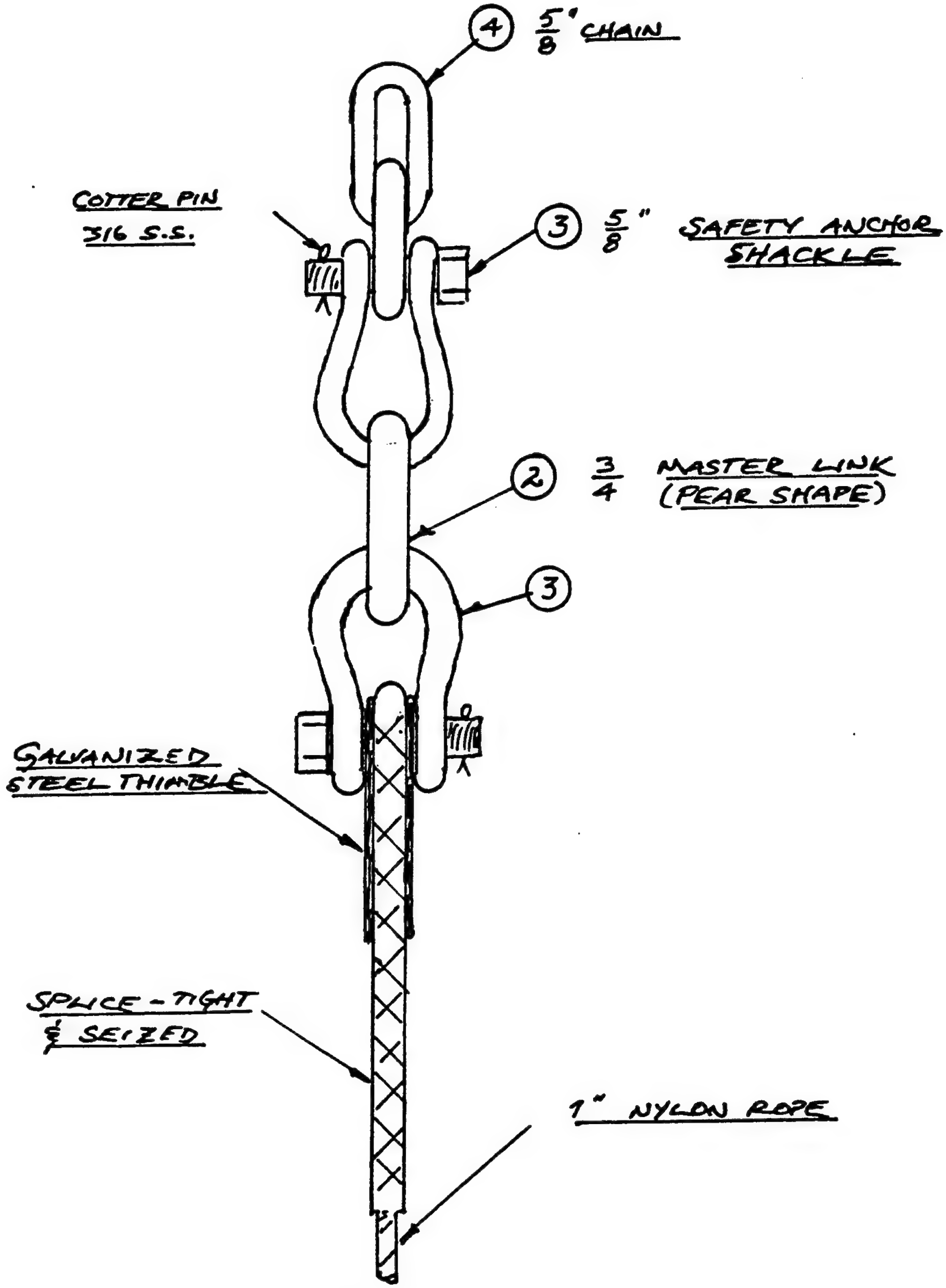
4. Connecting Hardware

The connecting hardware consists of galvanized, drop-forged steel shackles and sling (or master) links. A typical connection is shown in Figure 7.6. Sling links are inserted to ease deployment and recovery of the mooring.
A 17-inch glass ball float which has a buoyant force of about 50 lbs. can be secured to the 1-inch Nylon line at its midpoint to prevent the Nylon line on the downstream mooring from piling up on the sea floor and possibly becoming fouled.

The location of each of the components is shown in Figure 7.5 and the Bill of Materials for the mooring system is shown in Table 7.4. The number required, shown in Table 7.4, is for one complete mooring line and anchor. Generally, of course, the AMBP will require two complete mooring assemblies, one for each end of the hull. Thus, the total number required will be double that shown.

TABLE 7.4
BILL OF MATERIALS FOR THE MOORING SYSTEM

PART #	NUMBER REQ'D	NOMINAL SIZE	DESCRIPTION	COMMENTS
1	1	3/4-inch	Safety Anchor Shackle	Crosby G-2130 (7)
2	4	3/4-inch	Sling Link	Crosby G-341
3	5	5/8-inch	Safety Anchor Shackle	Crosby G-2130
4	1	5/8-inch	Proof Coil, Welded Chain	Crosby Grade 3
5	1	1-inch	Plaited or Braided Nylon Rope	
6	2	3/4-inch	Safety Chain Shackle	Crosby G-2150
7	1	1-inch	Proof Coil, Welded Chain	
8	1	100 lb.	NAVMOOR Anchor	Two Req'd for 5 knots
9	1	1000 lb.	Clump	Req'd for 5 knots
10	1	17-inch	Glass Ball Float or equiv.	Optional



Typical Mooring

System Connection

Figure 7.6

8. CONTROL SYSTEM DEVELOPMENT

The overall objective of the control system development tasks was to obtain the major control hardware and software, become familiar with its setup and programming, and to simulate a number of the control and display operations on the benchtop, including base station to mobile station communication using the radio modems. The hardware and software will then later be used in the construction of the prototype AMBP system.

8.1 HARDWARE

The principal test hardware obtained for the test and later use in the prototype AMBP system included the following:

OPTO 22 M4RTU Remote Telemetry Unit
OPTO 22 M4RTUX Input/Output Extender
OPTO 22 M4PS24D 24 VDC Power Supply
Various typical OPTO 22 I/O modules
OPTO 22 G4LC32ISA controller
Generic Pentium PC for use as a base station unit
Two Pacific Crest RFM96 radio modems with 15watt amplifiers

The M4RTUX and G4LC32ISA units were not used for the lboratory work but are available for use in an AMBP prototype.

8.2 SOFTWARE

Two OPTO 22 software packages were purchased for use with the OPTO Mistic controllers:

Cyrano 200
Mistic MMI

The Cyrano software package allows the programming of OPTO 22 Mistic controllers through a PC interface. The control functions are generated by drawing control charts using chart-based commands.

The Mistic MMI package allows the user to develop graphic screen interfaces for the PC control station. It is closely integrated with the Cyrano control software.

8.3 SYSTEM SIMULATION

The Cyrano and Mistic MMI software packages were used to program and develop an AMBP simulator to obtain experience with the software and serve as a basis for benchtop and modem testing.

Since an engine-pump system was not available to control, the OPTO 22 controller, itself, was employed to simulate the engine-pump system pressure-flow characteristics coupled with a time lag to represent the dynamics of the system. The program was formulated to allow the operator, through the PC interface, to arbitrarily change the desired engine speed and/or inlet pressure to the

pump. The operator can also set prescribed limits on minimum inlet pressure, maximum outlet pressure, and maximum engine speed.

The control system acting through the simulated pump system can vary or limit the engine speed to ensure that the pump inlet pressure does not drop below the set minimum value or that the pump outlet pressure does not rise above the set maximum value.

A Mistic MMI computer screen was programmed to show the variables both numerically and graphically and to allow the operator to set the various limits and input values using the computer keyboard and mouse. This screen is shown in Figure 8.1.

The Cyrano pump simulation program was printed out directly from the computer and is reproduced in Appendix A.

The program operated as intended. If the inlet pressure was set too high, simulating the placement of the shipboard pump too close to the AMBP, then the AMBP outlet pressure at the set RPM would tend to increase above the allowable outlet pressure. In this case, the controller automatically reduces the actual engine speed to hold the outlet pressure at the limit.

If the inlet pressure is decreased, the outlet pressure tries to decrease and the control system acts to increase engine speed to hold the outlet pressure at the set value. If, when the desired or set engine speed is increased, the inlet pressure tries to decrease below a set value at which cavitation or hose collapse could result, the control system will not allow the engine speed to further increase.

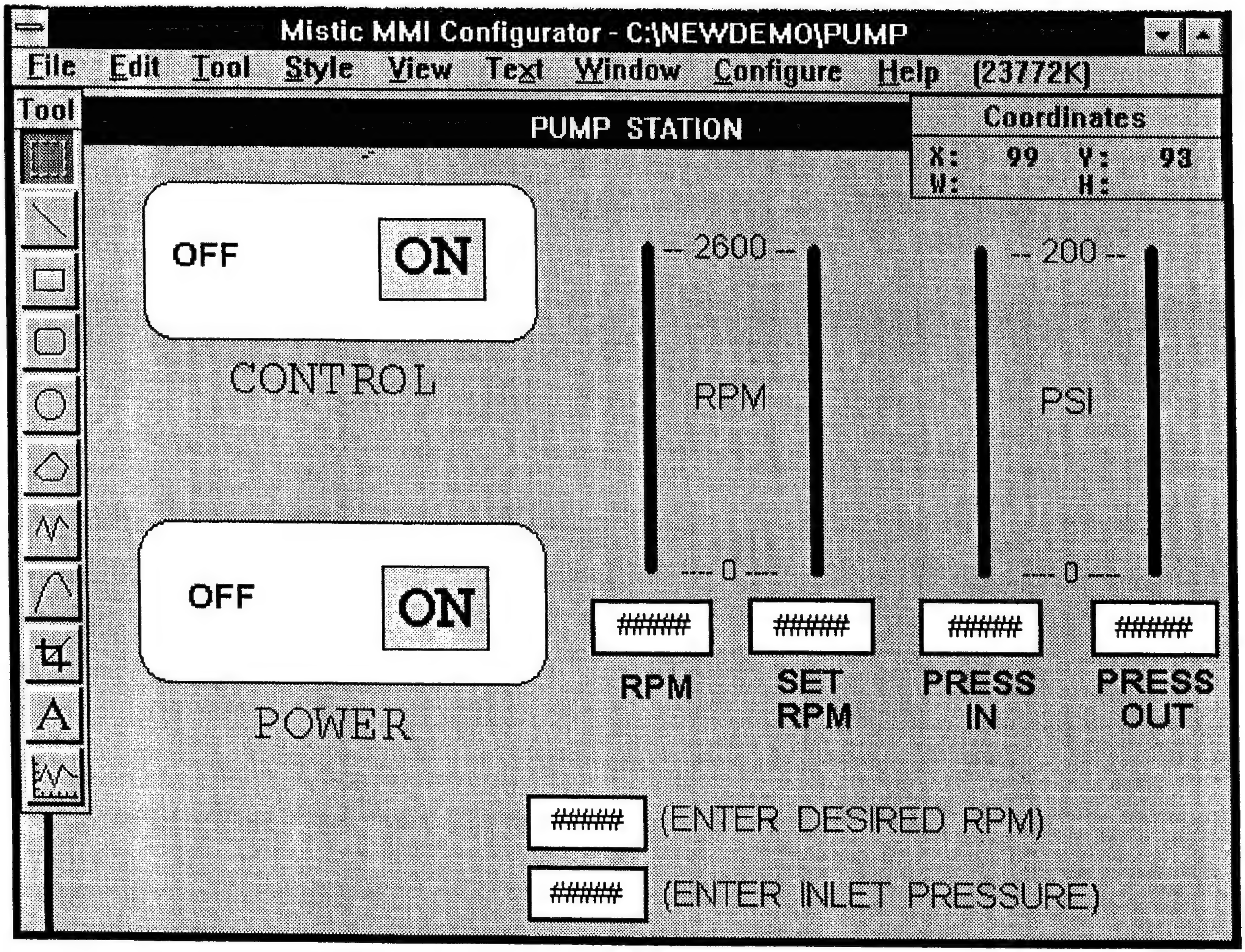
This simulation does not duplicate the actual system in includes a primary or shipboard pump, pipeline on both sides of the AMBP, and a load at the end of the pipeline. However, simulation of all of these systems would be extremely complex and would not further the objective of this program to obtain experience in programming the OPTO 22 software for the AMBP controller and demonstrating its function. The principle difference between the simulation control system and an actual control system is that the inlet pressure is an input for the simulation system whereas it is simply a measured quantity for the actual system.

8.4 DISPLAY OF VARIABLES

The Mistic MMI software allows a variety of methods for displaying measured quantities and control inputs. The screen shown in Figure 8.1 is an example of this type of display. That display, of course, is of use primarily for the simulation and range tests and is not intended as a prototype for the display which will be used in the final system.

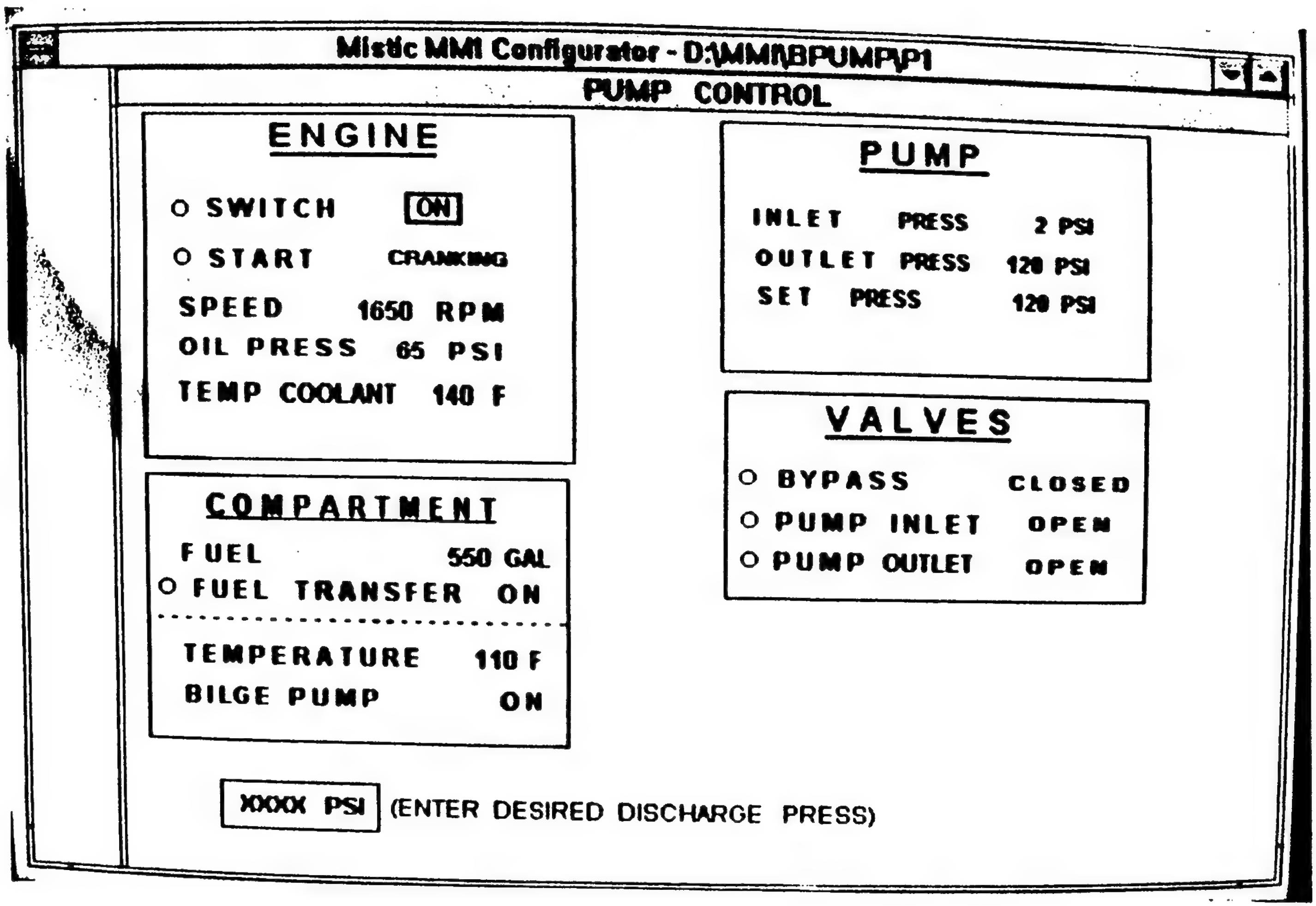
Although the display software is quite powerful and allows graphical depiction of switches, gauges, etc., it is perhaps desirable, given the amount of monitoring data necessary for the AMBP, to display the data simply in relevant groups. Since the primary settings are numerical values, it is perhaps desirable for the display to be essentially digital rather than analog in nature.

A preliminary version was programmed as shown in Figure 8.2, which is an actual photograph of the computer screen. For this display, the engine, pump, engine compartment, and flow valve variables are grouped for easy location.



Computer Screen for the Pump Simulation Program

Figure 8.1



Tentative Mistic MMI
Base Station Display

Figure 8.2

Engine variables include the "on-off" and starting switches which parallel their equivalents on the actual engine control panel. The remote versions would be controllable with the mouse on the base-station computer. In addition, digital readouts of engine RPM, oil pressure, and coolant temperature are given for easy reference in the same grouping. For the latter two quantities, an audible alarm can be sounded if the variable exceeds a set warning level and the engine could be shut off if they exceed a set absolute limit.

For the compartment, the only controllable variable is the fuel transfer valve which is controlled with the computer mouse. The other variables are simply displayed.

The pump inlet, outlet, and maximum outlet pressures are displayed in the pump grouping. The maximum discharge pressure is set at a separate point since it is the primary input control quantity. The limiting inlet pressure is not expected to change in the field and, thus, it is suggested this simply be programmed into the control.

In the valve grouping, the control is simply an on-off toggle for each valve controlled by the computer mouse. The open-closed indication would be derived directly from the limit switches on the valve, thus, showing the actual condition. If a hand-operated valve is used in the pump outlet, it would not be shown on the display.

The screen shown in Figure 8.2 is, of course, only tentative and will likely be refined later as the development of the prototype AMBP continues.

8.5 MODEM OPERATION

The frequencies assigned by the Navy for the modem tests were 416.275 and 419.9 Mhz. These frequencies were programmed into two of the available 15 frequency channels on the Pacific Crest radio modems.

The pump control simulation program described in Section 8.3 and Appendix A was used for all of the modem tests.

Initially, the M4RTU controller and the base station PC were connected directly with a cable until reliable communication could be established. Then, the modems were connected to each unit and tested until reliable communication could be established at short range in the laboratory.

Finally, the M4RTU was set up as a mobile unit, powered with batteries, and range tests were conducted to demostrate the operation of the system over a range consistent with its intended use.

(a) Settings

Considerable effort was required to achieve successful communication of data, first directly between the OPTO 22 unit and the base station computer and, then, with the modems inserted into the loop. Many variables and settings on the various pieces of the system were found critical to getting the system to operate. These are summarized as follows for future reference:

(1) Modem Mode of Operation - Several of the operating modes provided for the modems were tried, but the "Transparent mode EOT character" was found to be successful. The EOT (end of transmission) character was set to be "13" for "carriage return". It turns out that this mode of

operation and character are appropriate for ASCII character transmission and for compatibility with the OPTO 22 unit.

(2) Antenna - The modems came with short rubber-covered whip antennas that are intended to connect directly onto the BNC coax connectors on the units. We were never successful at getting the units to communicate with each other using these antennas. Probably, the proximity of the antennas to fields generated by the equipment fooled the receivers into thinking that a signal was always present for them to receive and prevented any transmission. When we replaced these rubber antennas with vertical whips connected by cable and separated from the units by a few feet, we ceased having these problems.

(3) Digisquelch - This setting determines the sensitivity of the receiver. Settings of High, Moderate, and Low can be used. We used "Low" for all tests in the laboratory. However, we tested both "Low" and "High" for subsequent range tests to determine whether the "High" setting would increase the range. If "High" is used in the laboratory, there is a tendency, even at 2 watts output, to lock up the receivers.

(4) COMM port - The COMM port of the M4RTU is normally wired up for communication with another computer with a jumper that emulates a null modem connection. If the same input connections are used with a pair of modems inserted between the computers, then a NULL MODEM must be inserted into the line between the modem and the M4RTU to reverse the effect of the jumper and return to the normal modem setup.

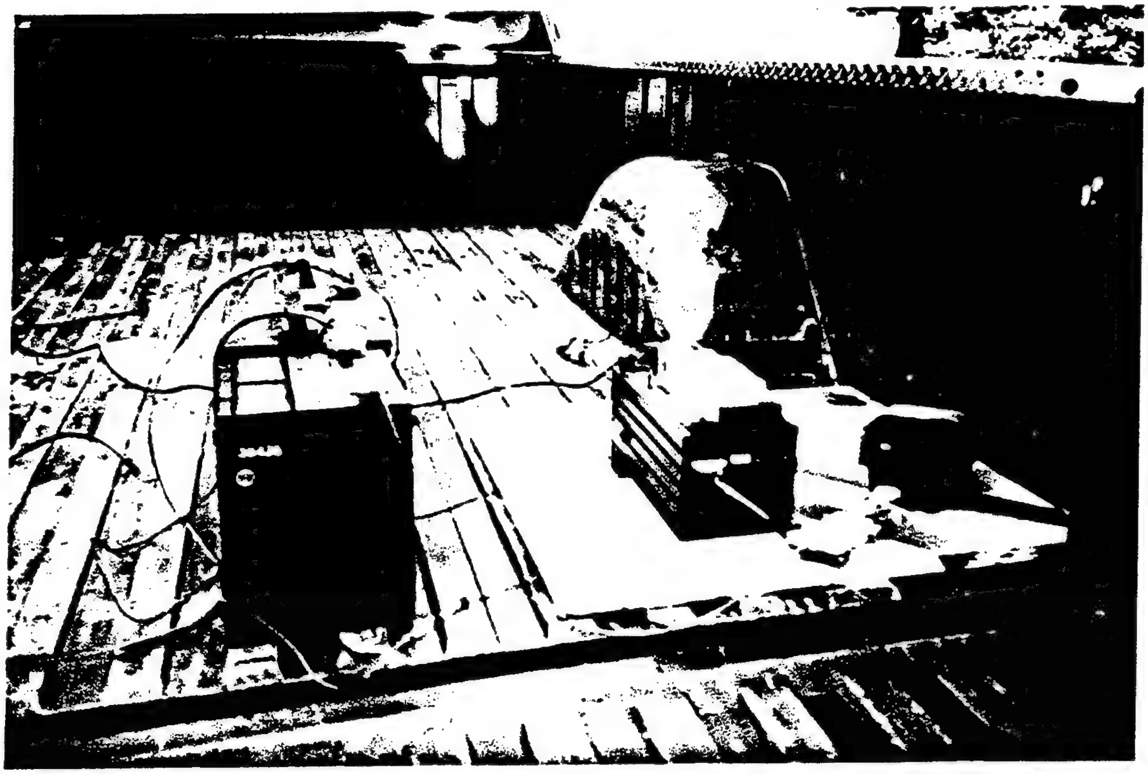
(5) Time Out Setting - The "time out" setting for the COMM port on the computer which is communicating with the M4RTU must be set at a value of at least 2 or 3 for successful communication with the M4RTU. This setting is made through the Mistic MMI Configurator on the base station computer.

With the above settings, we were able to achieve reliable communication between the M4RTU and the base station PC with the Cyrano and Mistic MMI software. As noted above, all tests were run using the pump control simulation program.

(b) Range Test

For the range test the Opto 22 M4RTU and one Pacific Crest modem were set-up in a truck for mobile operation. The units were powered by two 12 VDC automotive batteries. The Opto 22 unit requires 24 VDC. The modem requires 12 VDC which was supplied by a tap between the two batteries which were wired in series. The set-up is shown in Figure 8.3. The antenna used was a dual-band magnetic-mount whip for use on the 144 MHz. and 440 MHz. The modem was set for 416.275 MHz. which is one of the assigned Navy frequencies for the project.

The procedure for the tests was simple. The mobile system was driven out to various distances from the base station while the two operators communicated with each other via cellular phone to determine whether or not the M4RTU continued to communicate with the base station computer which was showing a display of the control system. The base station operator continued to vary the input speed control and inlet pressure setting to ensure that the base station was transmitting to the M4RTU as well as the reverse.

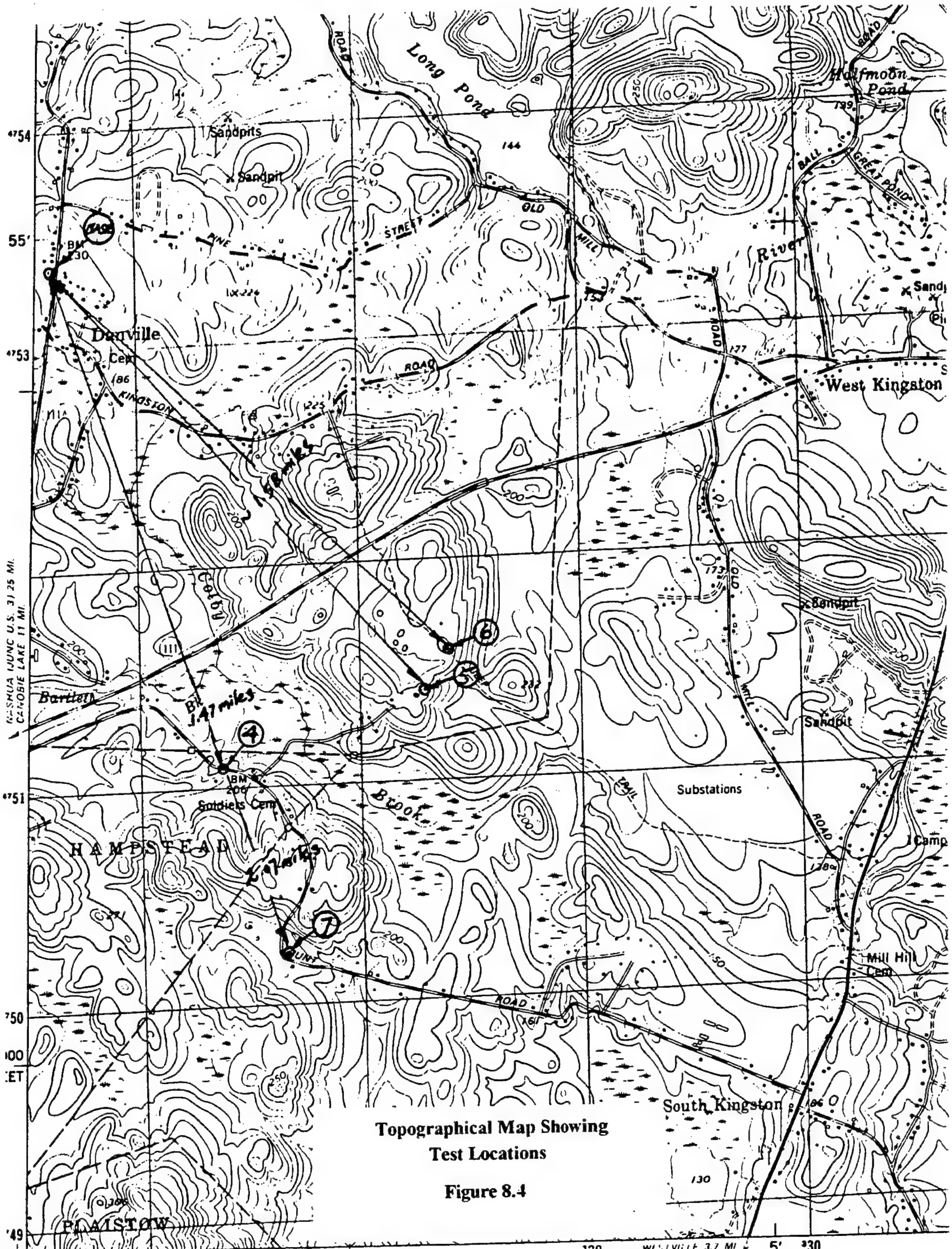


Photograph of the Mobile Set-Up
Figure 8.3

The results are summarized in Table 8.1. Map references for the longer range points are shown on the USGS topographical map of Figure 8.4.

Table 8.1
Results of the AMBP Communication Range Test

Receiver Sensitivity	Transmitter Power, watts	Range, Miles	Map Ref.	Contact Quality
Low	2	0.25	N/A	Good
		0.45	N/A	No Contact
	15	0.45	N/A	Good
	15	1.0	N/A	Good
	15	1.47	4	Intermittant
	15	1.58	6	No Contact
High	2	0.45	N/A	Good
	2	1.0	N/A	Good
	2	1.47	4	Good
	15	1.47	4	Good
	15	1.58	5	Good
	15	1.58	6	No Contact
	15	2.07	7	Good

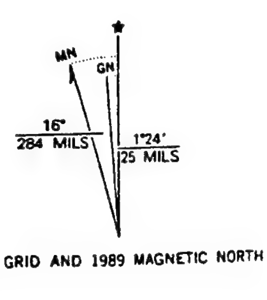


Topographical Map Showing Test Locations

Figure 8.4

Mapped, edited, and published by the Geological Survey

Control by USGS, NOS/NOAA, and New Hampshire Geological Survey



With low receiver sensitivity, the range was limited to about 1/4 mile with a transmitter power of 2 watts. For 15 watts, the range increases to about 1 mile.

For high receiver sensitivity, the range was about 2 miles with a power of 15 watts but no contact could be established beyond this range.

There is no question that the hilly terrain affected the attainable range considerably. At this frequency, communication is line-of-sight. As the mobile system moved away from the base station and passed behind small hills, it was noticed that communication was momentarily lost. Compare, for example, sites 5 and 6 of Figure 8.4 which are very close to each other. No communication, regardless of settings could be established at 6 which has a hill between it and the base station as shown on the map. However, the contact was strong at 5 which is the same distance but slightly off-set to avoid the hill.

The actual system requires communication over a range of about 2 miles, but from the beach to the moored AMBP directly off-shore. There will be no terrain problems for this set-up. It appears from the results of this test that there should be no problem in attaining the required range on the beach. It is quite likely that 2 watts will be ample under these conditions.

9. CONCLUSIONS

The principal conclusion that can be reached as a result of the further development studies conducted in Phase II of this program is that an AMBP, which meets the required performance specification, still appears feasible and no obstacles to the development of a reliable, practical field system have appeared.

An additional operational specification, the requirement to operate when left high and dry by the receding tide, has been found feasible. The recommended design uses the main pumped fuel flow as a source for cooling fluid for the engine. The prototype design developed during this phase, incorporates the hardware necessary to implement this feature.

The on-board electronic system requires a cabinet heater to operate at the coldest specified ambient temperature conditions. This system has been designed into the prototype electrical system. About 2 inches of thermal insulation will be required on the electronic cabinet during this very cold weather. It is also likely that a cold-weather package should be obtained for the engine if arctic operations are planned, and operating procedures, similar to other diesel-powered equipment, should be used. Requirements for continuous operation or periodic start-ups might be difficult for the AMBP, of course, if there is no fuel to pump for engine cooling.

Analysis shows that the temperature in the AMBP under the worst specified tropical conditions should be safe for the electronics and the engine. However, the actual local heat rejection from the engine is not known. Thus, tests should be run with the prototype system to refine the predicted compartment temperatures under the highest specified ambient conditions. Forced blower cooling of the compartment can always be used if necessary.

Bench testing of the remote control system hardware and software allowed the gaining of programming experience with the control chart-based software used to program the on-board computer. A simulated pump pressure and speed control program was written and used to test the controller and for range testing using the controller and radio modems.

The range testing, using the radio modems to provide communication between the on-board controller and a PC base terminal, showed that the range required on the beach during an amphibious operation should be no problem. A successful range of over 2 miles was obtained with testing in hilly terrain.

A set of drawings has been prepared for a prototype system which will allow full-scale field testing of the unit, and the machinery pallet portion of the system is currently being fabricated.

In order to produce a stable system, it was found necessary to increase the weight somewhat from that estimated in Phase I. The current estimated total weight of the AMBP is approximately 11,400 lbs. This is still within the capability of the handling equipment.

A mooring system was designed for the prototype AMBP which makes use of a single 100 lb. NAVMOOR anchor for a design current of 2.5 knots. Approximately 40 ft. of 1-inch chain is required, in addition, at the bottom end of the mooring in order to balance the vertical load..

At a survival current of 5 knots, a second NAVMOOR anchor is required for the horizontal load, and the addition of either a 1000 lb. clump anchor or the lengthening of the 1-inch chain from 40 ft. to approximately 160 ft. to balance the vertical load.

10. RECOMMENDATIONS

It is recommended that the detailed prototype design developed in the tasks reported in this report be fabricated and tested to verify the predicted performance or to illuminate design modifications, where necessary.

The most important subsystem is the machinery pallet subsystem which contains the on-board control system and all of the mechanical and hydraulic hardware. The prototype machinery pallet is currently under fabrication. When it is completed, the control system should be programmed to correctly control the engine and valves and display the various transducer signals. It is recommended that a special test pumping loop be set up which will allow pumping up to 600 GPM of fuel or water and that the system be tested and debugged. The engine can be tested without large throughput, if necessary, by providing an artificial flow of about 60 GPM to the primary of the engine heat exchanger. However, sufficient fluid must be supplied to the pump to keep its shaft seals lubricated.

When possible, the prototype AMBP hull assembly should be fabricated and assembled with the pallet assembly. The total system should then be tested for flotation characteristics.

Simulated maintenance procedures should be evaluated. In particular, the practicality of removing the equipment pallet each time significant maintenance operations are required must be assessed. Although the overall width of the AMBP cannot be increased while meeting the requirement that the system be transportable by ISO container, it is possible to increase the length of the AMBP. Thus, it would be possible to provide a small operating position in front of the electrical panel with a fairly low overhead. This would possibly allow manual operation of the system from the engine compartment and some electrical or electronic maintenance tasks.

Another set of tests for the complete system should be aimed at establishing the actual engine heat loads when operating at full power to allow a more accurate compartment temperature prediction for the high ambient temperature limit.

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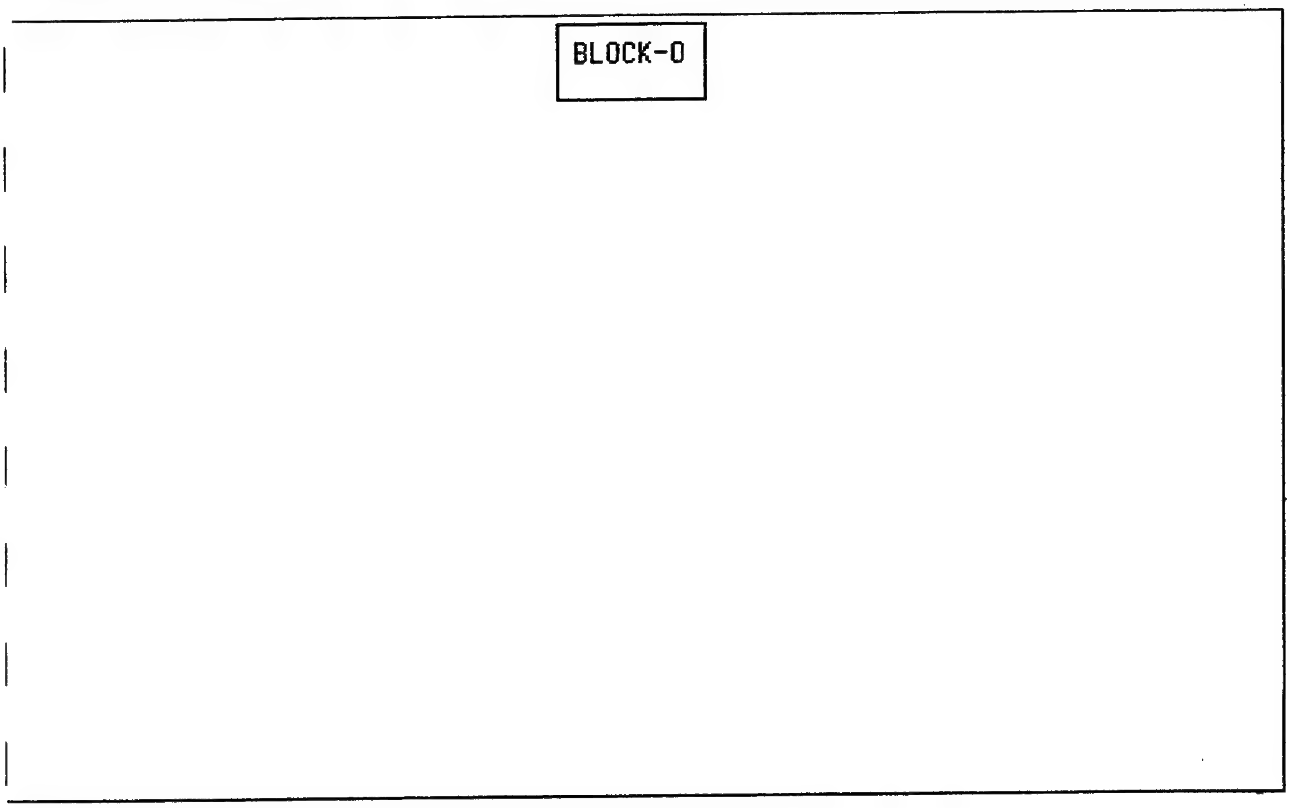
APPENDIX

CYRANO PUMP SIMULATION PROGRAM

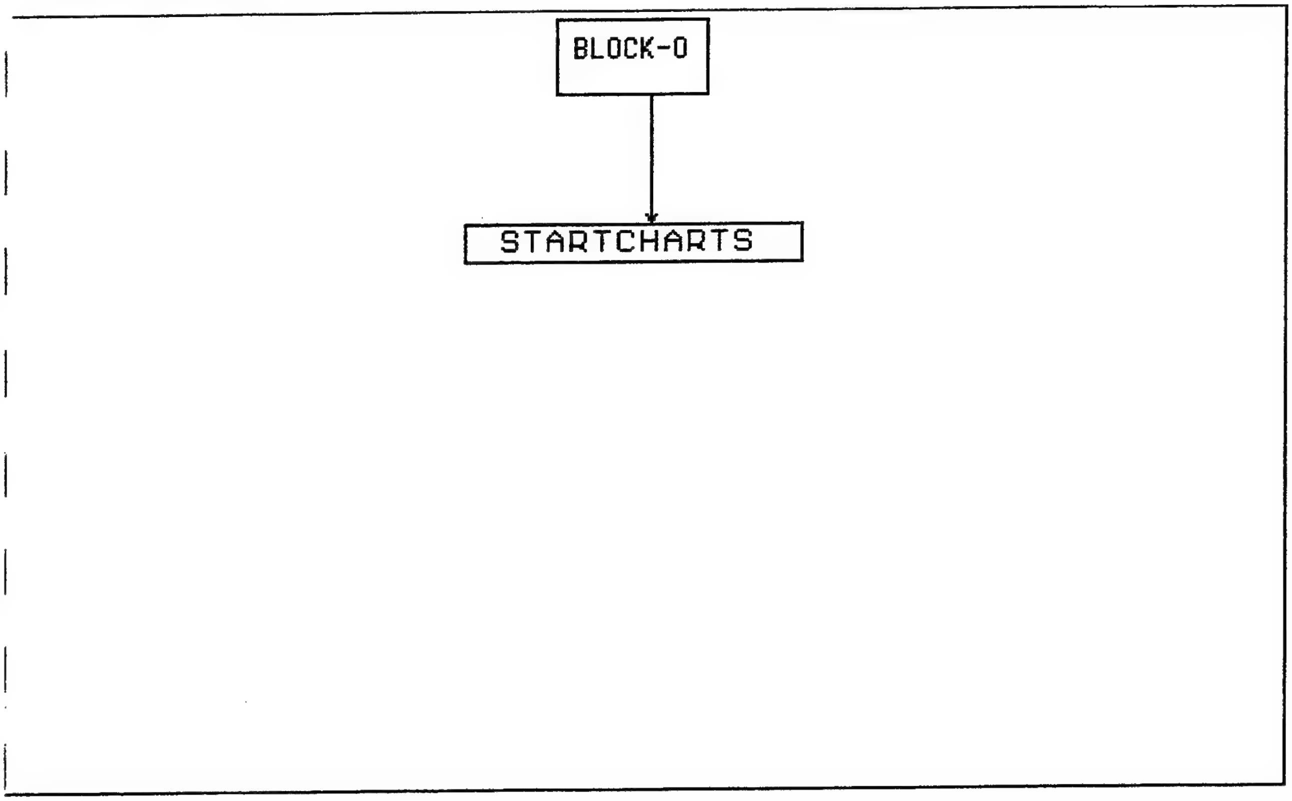
Chart Column: 1



(a) BLOCK-0 **Has No Exit**

No Condition Blocks in this chart.

Chart Column: 1



(a) BLOCK-0	**Exit To -> (a) STARTCHARTS**

(a) STARTCHARTS	**Has No Exit**
START CHART	PROC1
Put Status In	PROC1_CHART

No Condition Blocks in this chart.

Chart Column: 1

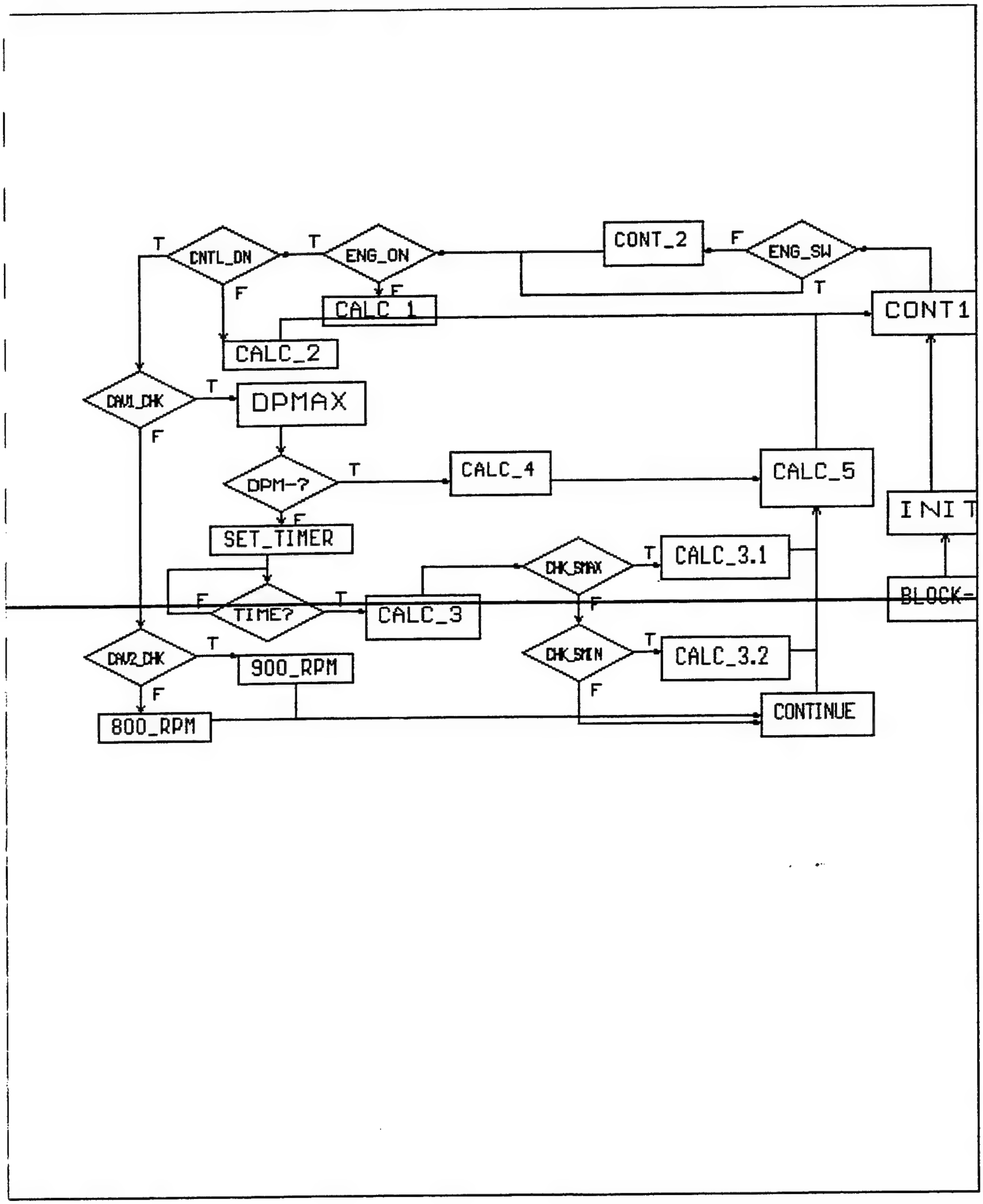
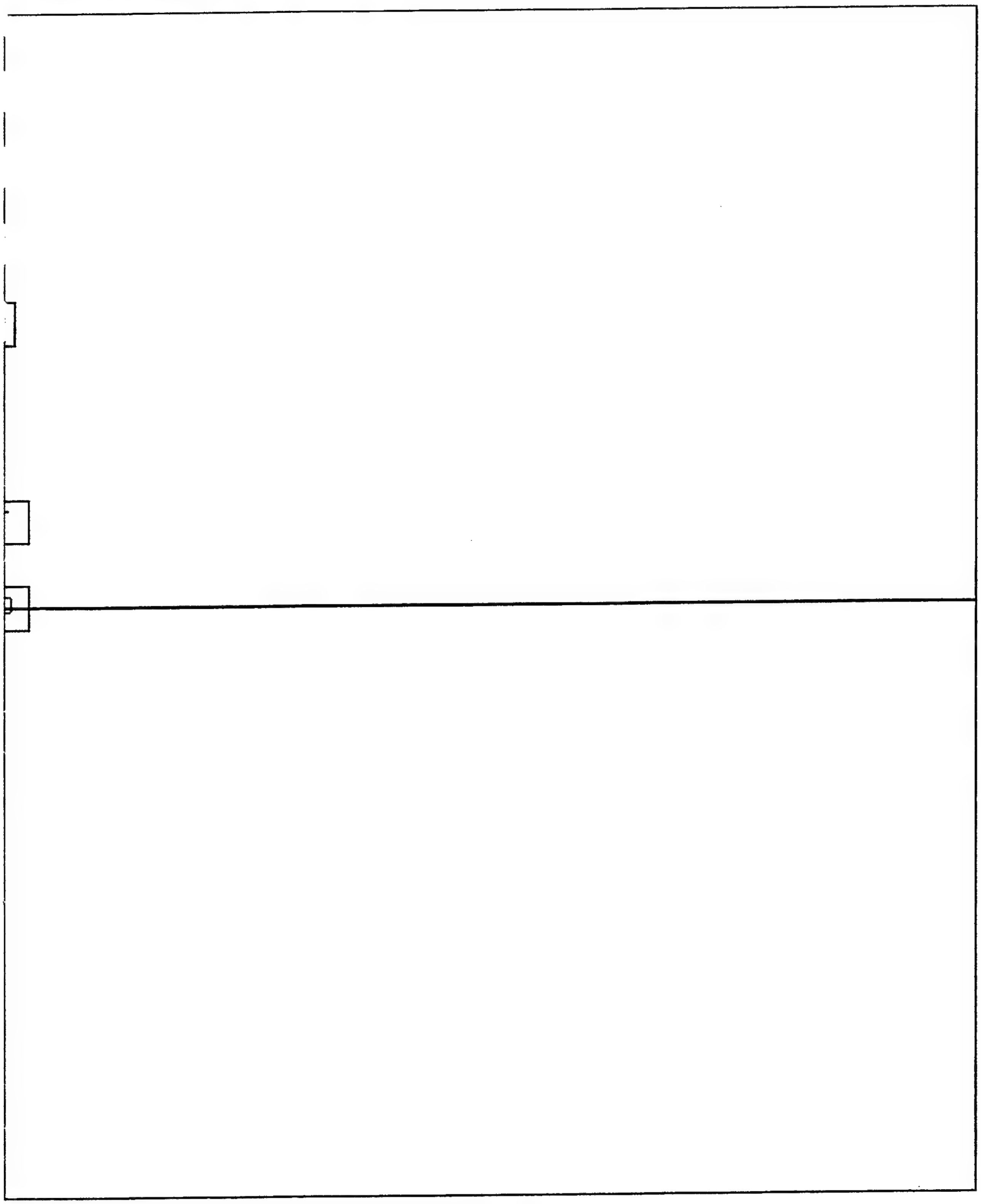


Chart Column: 2



(a) 800_RPM	**Exit To -> (a) CONTINUE**
MOVE	
From	800.0000
To	S

(a) 900_RPM	**Exit To -> (a) CONTINUE**
MOVE	
From	900.0000
To	S

(a) BLOCK-0	**Exit To -> (a) INIT**

(a) CALC_1	**Exit To -> (a) CONT1**
MOVE	
From	0
To	CNTL_SW
MOVE	
From	PIN
To	POUT

(a) CALC_2	**Exit To -> (a) CONT1**
DO MULTIPLY	
	S
Times	S
Put Result In	DP
DO MULTIPLY	
	DP
Times	C1
Put Result In	DP
DO DIVIDE	
	DP

By	C2
Put Result In	DP

DO ADDITION

	PIN
Plus	DP
Put Result In	POUT

(a) CALC_3 **Exit To -> (c) CHK_SMAX**

DO MULTIPLY

	DPMAX
Times	C2
Put Result In	S2

DO DIVIDE

	S2
By	C1
Put Result In	S2

TAKE SQUARE ROOT OF

	S2
Put Result In	SMAX

DO SUBTRACTION

	SS
Minus	S
Put Result In	DS

DO DIVIDE

	DS
By	2.000000
Put Result In	DS

DO MULTIPLY

	DS
Times	1.010000
Put Result In	DS

DO ADDITION

	S
Plus	DS
Put Result In	S

MOVE

From	S
To	S_I

MOVE

From	S_I
To	S

(a) CALC_3.1 Exit To -> (a) CALC_5

MOVE

From	SMAX
To	S

(a) CALC_3.2 Exit To -> (a) CALC_5

MOVE

From	SMIN
To	S

(a) CALC_4 Exit To -> (a) CALC_5

MOVE

From	SMIN
To	S

(a) CALC_5 Exit To -> (a) CONT1

DO MULTIPLY

	S
Times	S
Put Result In	S2

DO MULTIPLY

	S2
Times	C1
Put Result In	DP

DO DIVIDE

	DP
By	C2
Put Result In	DP

DO ADDITION

	PIN
Plus	DP
Put Result In	POUT

(a) CONT1		**Exit To -> (c) ENG_SW**
MOVE		
	From	S
	To	S_I
MOVE		
	From	S_I
	To	S
MOVE		
	From	SS
	To	SS_I
MOVE		
	From	SS_I
	To	SS

(a) CONTINUE		**Exit To -> (a) CALC_5**
MOVE		
	From	S
	To	S

(a) CONT_2		**Exit To -> (c) ENG_ON**
MOVE		
	From	0
	To	CNTL_SW
MOVE		
	From	0.000000
	To	S

(a) DPMAX		**Exit To -> (c) DPM-?**
DO SUBTRACTION		
		PMAX
	Minus	PIN
	Put Result In	DPMAX

(a) INIT		**Exit To -> (a) CONT1**
MOVE		
	From	0
	To	ENG_SW
MOVE		
	From	0
	To	CNTL_SW
MOVE		
	From	1000.000
	To	S

(a) SET_TIMER		**Exit To -> (c) TIME?**
MOVE		
	From	2.000000
	To	TIMER

(c) CAV1_CHK		TRUE Exit To -> (a) DPMAX FALSE Exit To -> (c) CAV2_CHK
GREATER OR EQUAL		
	Is	PIN
	To	PCAV1

(c) CAV2_CHK	TRUE Exit To -> (a) 900_RPM FALSE Exit To -> (a) 800_RPM

(c) CHK_SMAX		TRUE Exit To -> (a) CALC_3.1 FALSE Exit To -> (c) CHK_SMIN
GREATER		
	Is	S
	Than	SMAX

(c) CHK_SMIN		TRUE Exit To -> (a) CALC_3.2 FALSE Exit To -> (a) CONTINUE
LESS		
	Is	S
	Than	SMIN

(c) CNTL_ON	TRUE Exit To -> (c) CAV1_CHK FALSE Exit To -> (a) CALC_2

(c) DPM-?		TRUE Exit To -> (a) CALC_4 FALSE Exit To -> (a) SET_TIMER
LESS		
	Is	DPMAX
	Than	0.000000

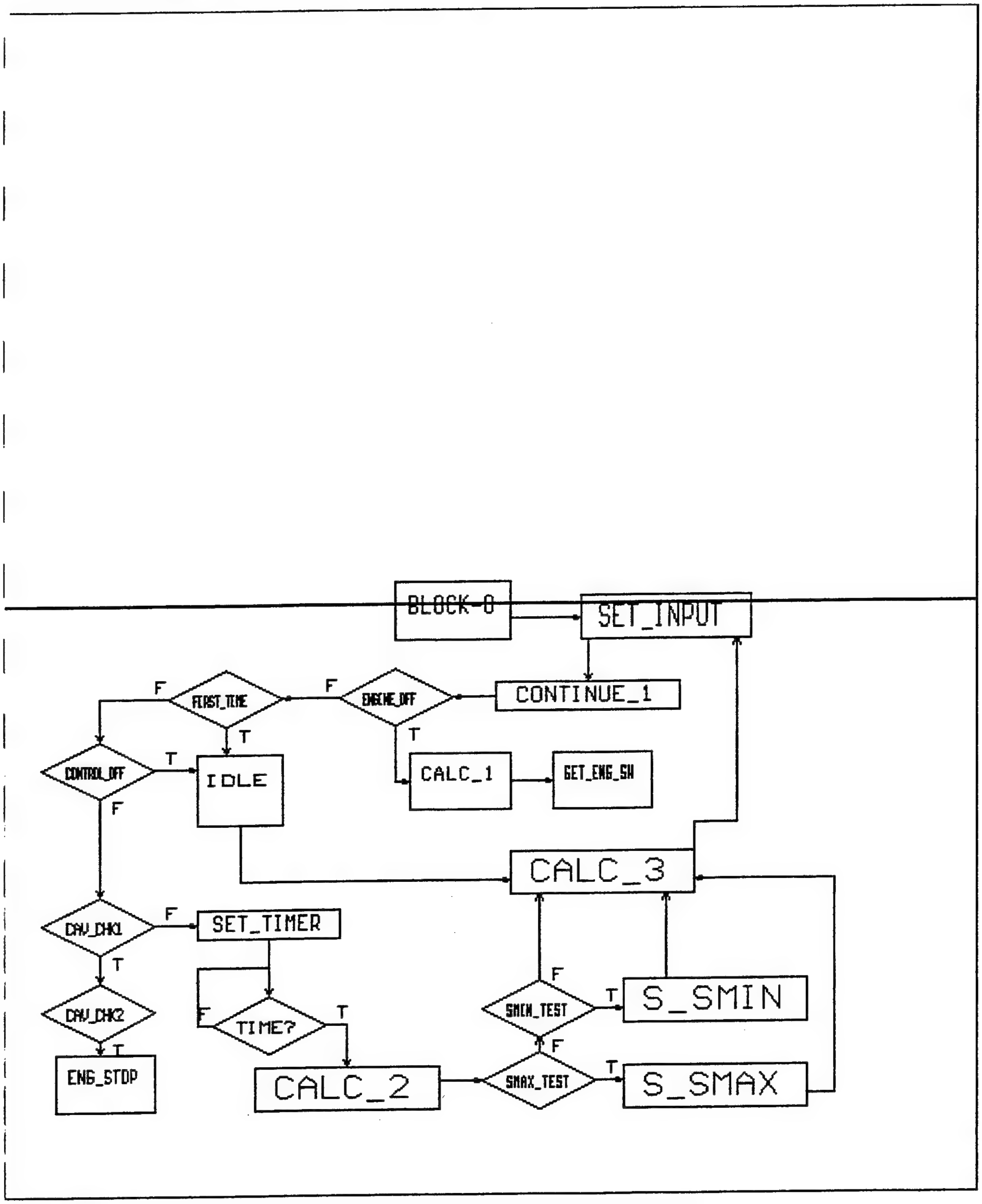
(c) ENG_ON		TRUE Exit To -> (c) CNTL_ON FALSE Exit To -> (a) CALC_1
EQUAL		
	Is	ENG_SW

```
        To                        0
-------------------------------------------------------------------
(c) ENG_SW                        TRUE Exit To ->  (c) ENG_ON
                                  FALSE Exit To -> (a) CONT_2

    EQUAL
        Is                        ENG_SW
        To                        0
-------------------------------------------------------------------
(c) TIME?                         TRUE Exit To ->  (a) CALC_3
                                  FALSE Exit To -> (c) TIME?

    TIMER EXPIRED?
        Is                        TIMER
-------------------------------------------------------------------
```

Chart Column: 1



(a) BLOCK-0 **Exit To -> (a) SET_INPUT**

(a) CALC_1 **Exit To -> (a) GET_ENG_SW**

MOVE

From	PIN
To	POUT

(a) CALC_2 **Exit To -> (c) SMAX_TEST**

DO SUBTRACTION

	SS
Minus	S
Put Result In	DS

DO MULTIPLY

	DS
Times	0.5000000
Put Result In	DS

DO ADDITION

	S
Plus	DS
Put Result In	S

(a) CALC_3 **Exit To -> (a) SET_INPUT**

DO MULTIPLY

	S
Times	S
Put Result In	DP

DO MULTIPLY

	DP
Times	C1
Put Result In	DP

DO DIVIDE

	DP
By	100000.0
Put Result In	DP

```
    DO ADDITION
                                  PIN
        Plus                      DP
        Put Result In             POUT
```

```
(a) CONTINUE_1                    Exit To -> (c) ENGINE_OFF
```

```
(a) ENG_STOP                      Has No Exit
```

```
(a) GET_ENG_SW                    Has No Exit
```

```
(a) IDLE                          Exit To -> (a) CALC_3

    MOVE
        From                      SIDLE
        To                        S

    DO MULTIPLY
                                  S
        Times                     S
        Put Result In             DP

    DO MULTIPLY
                                  DP
        Times                     C1
        Put Result In             DP

    DO DIVIDE
                                  DP
        By                        100000.0
        Put Result In             DP

    DO ADDITION
                                  PIN
        Plus                      DP
        Put Result In             POUT
```

```
(a) SET_INPUT                     Exit To -> (a) CONTINUE_1
```

MOVE		
	From	40.00000
	To	PIN
MOVE		
	From	1500.000
	To	SS
MOVE		
	From	-1
	To	ENG_SW
MOVE		
	From	-1
	To	CNTL_SW
DO SUBTRACTION		
		PMAX
	Minus	PIN
	Put Result In	DPMAX
DO DIVIDE		
		DPMAX
	By	C1
	Put Result In	S2
DO MULTIPLY		
		S2
	Times	100000.0
	Put Result In	S2
TAKE SQUARE ROOT OF		
		S2
	Put Result In	SMAX

(a) SET_TIMER **Exit To -> (c) TIME?**

MOVE		
	From	2.000000
	To	TIMER

(a) S_SMAX **Exit To -> (a) CALC_3**

MOVE		
	From	SMAX
	To	S

(a) S_SMIN **Exit To -> (a) CALC_3**

(c) CAV_CHK1		TRUE Exit To -> (c) CAV_CHK2 FALSE Exit To -> (a) SET_TIMER
LESS		
	Is	PIN
	Than	10.00000

(c) CAV_CHK2	TRUE Exit To -> (a) ENG_STOP No FALSE Exit

(c) CONTROL_OFF	TRUE Exit To -> (a) IDLE FALSE Exit To -> (c) CAV_CHK1

(c) ENGINE_OFF		TRUE Exit To -> (a) CALC_1 FALSE Exit To -> (c) FIRST_TIME
EQUAL		
	Is	ENG_SW
	To	0

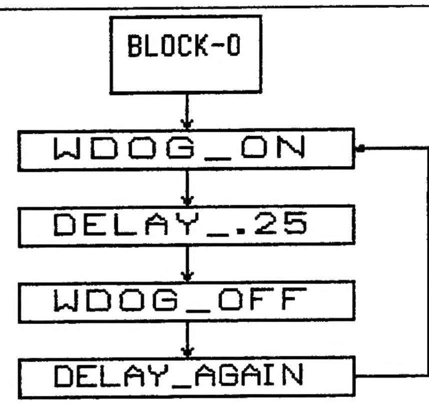
(c) FIRST_TIME		TRUE Exit To -> (a) IDLE FALSE Exit To -> (c) CONTROL_OFF
EQUAL		
	Is	FIRST_TIME
	To	0

(c) SMAX_TEST		TRUE Exit To -> (a) S_SMAX FALSE Exit To -> (c) SMIN_TEST
GREATER		
	Is	S
	Than	SMAX

(c) SMIN_TEST	TRUE Exit To -> (a) S_SMIN FALSE Exit To -> (a) CALC_3

(c) TIME?	**TRUE Exit To -> (a) CALC_2** **FALSE Exit To -> (c) TIME?**
TIMER EXPIRED?	
Is	TIMER

Chart Column: 1



(a) BLOCK-0	Exit To -> (a) WDOG_ON
(a) DELAY_.25	Exit To -> (a) WDOG_OFF
DELAY (MSEC)	DOGTIME
(a) DELAY_AGAIN	Exit To -> (a) WDOG_ON
DELAY (MSEC)	DOGTIME
(a) WDOG_OFF	Exit To -> (a) DELAY_AGAIN
TURN OFF	WDOG
(a) WDOG_ON	Exit To -> (a) DELAY_.25
TURN ON	WDOG

No Condition Blocks in this chart.